AD-A079 291 SHIP STRUCTURE COMMITTEE WASHINGTON DC F/G 13/10
REPORT ON SHIP VIBRATION SYMPOSIUM 1978, SHERATON NATIONAL HOTE--ETC(U)
SEP 79 E S DILLON
UNCLASSIFIED SSC-292 NL

END
DATE
FILMED
3 80

Member Agencies:
United States Coast Guard
Naval Sea Systems Command
Military Sealift Command
Maritime Administration
United States Geological Survey
American Bureau of Shipping

Ship Structure Committee

An Interagency Advisory Committee
Dedicated to Improving the Structure of Ships

Address Correspondence to:
Secretary, Ship Structure Committee
U.S. Coast Guard Headquarters,(G-M/82)
Washington, D.C. 20590

September 1979

During the preliminary period of organizing the Ship Vibration Symposium, for October 16-17, 1978, sponsored jointly by the Ship Structure Committee and The Society of Naval Architects and Marine Engineers, the papers and discussions were viewed as an opportunity to assess the state-of-the-art of vibration technology and to document areas where further work may be needed. The Ship Structure Committee, therefore, requested the Symposium Steering Committee to have a post-mortem report prepared that would summarize key conclusions and outline recommendations for future research work.

This document constitutes that post-mortem report.

Henry H. Bell
Henry H. Bell
Rear Admiral, U.S. Coast Guard
Chairman, Ship Structure Committee

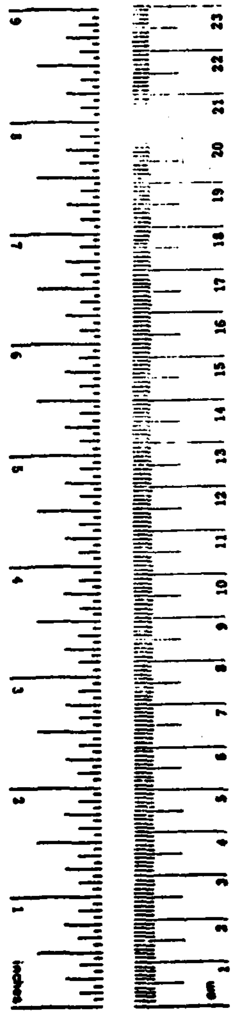
Justification
By
Distribution/
Availability Codes
Dist | Avail and/or special
A

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

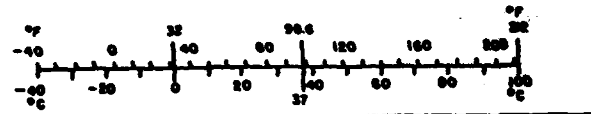
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in ≡ 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



Technical Report Documentation Page

1. Report No. SSC-292	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle REPORT ON SHIP VIBRATION SYMPOSIUM '78, SHERATON NATIONAL HOTEL, ARLINGTON, VA, October 16, 17, 1978		5. Report Date September 1979
		6. Performing Organization Code
7. Author(s) E. SCOTT DILLON		8. Performing Organization Report No. 60
9. Performing Organization Name and Address E. Scott Dillon, Consultant, 9009 Linton Street, Silver Spring, MD 20901		10. Work Unit No. (TRAIS)
		11. Contract or Grant No.
12. Sponsoring Agency Name and Address Commandant, Office of Merchant Marine Safety, U.S. Coast Guard Headquarters, Washington, D.C. 20590		13. Type of Report and Period Covered FINAL REPORT,
		14. Sponsoring Agency Code G-M/TP24

15. Supplementary Notes

16. Abstract

This report summarizes key conclusions and recommendations reached at Ship Vibration Symposium 1978.

17. Key Words		18. Distribution Statement
Propellers	Structural Analysis	Document is available to the public through the National Technical Information Service, Springfield, VA 22161
Vibration	Hydrodynamic Forces	
Ship Hull Structures	Cavitation	

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
UNCLASSIFIED	UNCLASSIFED	51	–

Form DOT F 1700.7 (8-72) Reproduction of completed page authorized

322 350

FORWARD

This report summarizes key conclusions and recommendations reached at Ship Vibration Symposium '78, a symposium dedicated to the memory and accomplishments of the late Professor Frank M. Lewis and sponsored jointly by the interagency Ship Structure Committee and the Hull Structure Committee of The Society of Naval Architects and Marine Engineers.

During the intense two-day period, October 16-17, 1978, an international group of 292 participants representing shipbuilders, ship designers, ship owners, researchers, classification and governmental organizations gathered and discussed all aspects of shipboard vibration, noise and hull/machinery compatibility. The eighteen technical papers presented at Ship Vibration Symposium '78 are contained in the printed proceedings and the formal discussion and authors' closures are available as a set of two volumes from The Society of Naval Architects and Marine Engineers. The purpose of this third printed volume is to summarize key conclusions and recommendations reached at the symposium.

Since the topics of "vibration and noise" are complex and not fully mastered, the reader will note a certain amount of controversy and conflicting views and recommendations outlined in the report. This situation, however, reflects in as accurate manner as possible, the actual written and verbal discussion that took place at the two-day symposium. In fact, in order to ensure as complete and accurate a report as possible, the draft version of the report was distributed and reviewed by the four principal SNAME panels involved with vibration and noise namely: Panel HS-7 (Vibrations), Panel H-8 (Unsteady Propeller Hydrodynamics), Panel M-20 (Machinery Vibrations), and Panel M-27 (Machinery Noise). Thus this report also reflects the comments of these four Technical & Research panels.

Finally, it should be brought to the readers attention that the three printed documents, namely: (1) the proceedings, (2) the discussion and authors' closure volume, and lastly, (3) this summary report, collectively assess the state-of-the-art of the broad subject of "vibration and noise". The primary purpose and main objective of this report, however, is unique in that it focuses in on the question "where are we now and where should we be headed"? Thus, this report is a key planning document that will serve as a basic reference for the next five to ten years.

N. O. HAMMER
Chairman
Ship Vibration Symposium '78

CONTENTS

PAGE

Forward v

Part I Summary 1

Part II Introduction and tabulations of complaints 5

Part III Vibration 9

General 9

Propeller-induced vibration 10

Source strength reduction 10

Wake optimization 10

Propeller blade cavitation minimization and blade skew 11

Prediction of propeller-induced forces and moments 12

Hull pressure forces 13

Bearing force and moment predictions 16

Unsteady forces acting on the rudder 18

Hull structure response 18

Added mass 20

Damping 21

Machinery system longitudinal vibration 21

Machinery system lateral vibration 22

Machinery system torsional vibration 22

Hull machinery compatibility 23

Wave-induced hull vibration - springing 23

Vibration signature analysis as a preventative maintenance tool aboard ship 24

Conclusions 26

General 26

Excitation - propeller forces - general 28

Propeller surface force excitation 30

Propeller bearing force excitation 31

Wave excitation 32

Response - general 32

Hull response 33

Machinery response 35

Vibration Signature Analysis for Preventative Maintenance 36

Recommendations 37

Part IV Noise 42

Discussion 42

Conclusions 47

Recommendations 50

SYMPOSIUM PROCEEDINGS

CONTENTS

State of the Art for Shipboard Vibration and Noise Control
E. F. Noonan and *S. Feldman* .... A

Vibration from a Ship Owner's Standpoint
R. McFarland and *D. Lindquist* .... B

The Considerations of Vibrations and Noise at the Preliminary and Contract Levels of Ship Design
N. M. Maniar and *J. C. Daidola* C

Vibration from a Shipbuilder's Point of View
R. D. Glasfeld and *D. C. MacMillan* .... D

Noise and Vibration as Viewed by a Maritime Union
R. F. Schamann .... E

Cost Considerations in Ship Vibration and Noise Problems
F. J. Dashnaw .... F

Approaches to Noise Exposure
S. E. Wehr .... G

Some Hydrodynamic Considerations of Propeller Induced Ship Vibrations
S. Hylarides .... H

Calculation and Comparison of Propeller Unsteady Pressure Forces on Ships
W. S. Vorus, J. P. Breslin, and *Y. S. Tien* I

Interaction and Compatibility Between Machinery and Hull from a Static and Vibratory Point of View
G. C. Volcy .... J

Computer Techniques for Use in Ship Hull Vibration Analysis and Design
F. E. Reed and *O. H. Burnside* .... K

Propeller and Wave-Induced Hull Structure Vibrations
S. G. Stiansen L

Dynamic Analysis of the Afterbody of a Ship—Towards a Successful Correlation Between Analytical and Experimental Results
J. L. Armand, P. Orsero, and *O. Robert* .... M

Noise Prediction and Prevention in Ships
A. C. Nilsson .... N

OSHA Noise Levels and the Marine Industry
F. A. Thoma .... O

Vibration Signature Analysis as a Preventive Maintenance Tool Aboard Ship
J. B. Catlin, Jr. .... P

Systematic Experiments to Determine the Influence of Skew and Rake on Hull Vibratory Excitation Due to Transient Cavitation
J. E. Kerwin, S. D. *Lewis* and S. *Kobayashi* .... Q

Highly Skewed Propellers—Full Scale Vibration Test Results and Economic Considerations
N. O. Hammer and *R. F. McGinn* .... R

PART I SUMMARY

Anyone believing that modern technology has solved shipboard vibration and noise problems would have second thoughts upon reading the long list of complaints from shipowners and maritime labor as revealed in papers "B" and "C" delivered in this Symposium. Far from pertaining to ancient vessels, this fervent criticism was leveled against recently delivered ships. One must conclude that either design knowledge is lacking or that it is not sufficiently understood to be readily applied to the average ship construction situation. It was the purpose of this Symposium to summarize results of ongoing research with the view to recording an incremental advance in the state of the art. It also pointed the way for much needed further research and development.

As can be deduced from conflicting conclusions and recommendations discussed in the text of the main body of this report and as listed at the ends of Parts III and IV, the "experts" disagree or admit lack of knowledge concerning a number of important aspects. Clearly a major long-range effort is still required to fully understand underlying phenomena and provide design tools for rational design so as to make possible with assurance the elimination of objectionable vibration and noise in future ships.

VIBRATION

As is well known, ship propellers rotate in a wake which includes portions of an entrained water boundary layer. Its variable forward speed is generated from friction along the length of the travelling hull underwater body. The effect of suction behind the stern due to propeller action together with possible separation as water fills the volume displaced by the passing ship, all together compound the wake in a pattern defying theoretical prediction of its variable velocities. Consequently, during its 360 degree rotation, each propeller blade encounters changing angles of attack which repeat with each rotation. Thus, the combined effects of all blades of a multibladed propeller produce alternating thrust, torque, and bearing forces and moments as well as pressure fields acting against the ship stern at propeller-blade frequency and multiples of this frequency.

Whereas overloaded propellers cavitate in uniform flow, the inception of cavitation will occur at much lower power absorption in the ship's uneven wake. This phenomenon greatly augments propeller forces acting on the ship. Since the forces and moments repeat at blade frequency, responses occur within the ship in the form of vibration and noise. Because of the elastic properties of a ship, the vibratory response amplifies to much higher levels when there is resonance with either the natural frequency of the entire hull, or if locally, of the natural frequency of its component parts.

Other forces causing vibration pertain to unbalanced or misaligned machinery and wave encounter. However, propeller-induced vibration predominates as the originator of most of the vibration difficulties suffered in ship service. Although scientists have worked diligently, especially during the last two decades, to better understand underlying phenomena, and indeed have made substantial progress; nevertheless, the more rigorous demands of higher powered

complex ships of recent vintage have enlarged the problem and put a higher price tag on the value of its solution.

Researchers achieved considerable success in calculating responses to known forces acting on the hull and propeller. However, these results are clouded by uncertainties as to amount and location of added mass from surrounding water and, in a resonant condition, the damping effect both hydrodynamical and structural, and damping from deadweight items of cargo, fuel oil, fresh water tankage, etc. Uncertainty also exists with respect to calculations of propeller-induced forces especially in a cavitating regime. The problem is compounded by the fact that the propeller distorts the wake in a manner most difficult to determine.

The quest for a quiet ship is further hampered in many cases by a compressed time schedule for ship delivery which compels design decisions before desired information on wake survey and structural arrangement become available. Even when model tests have been conducted, the state of the art is such that important discrepancies with respect to force measurements and cavitation bubble size exist as between model and full scale, and between calculations and model and full scale.

Inspite of these impediments, the Symposium developed general agreement that the application of present technology will improve prospects for success in achieving a quiet ship. However, attending shipbuilders were emphatically unwilling to guarantee attainment of specified vibration limitations because of uncertainties in the available design procedures.

Entirely apart from propeller-induced vibration, periodic wave encounters will sometimes synchronize with the natural longitudinal hull two nodal bending frequency of long slender ships such as Great Lakes ore carriers or high-speed destroyers. The resonating response known as "springing" induces significant stress in the longitudinal hull girder which combines with that emanating from uneven bouyancy in waves and dynamic effects. Because of the importance of this phenomenon, theoretical analysis together with full-scale measurements have been employed to develop rational treatment for design. Paper "L" reports on the considerable progress to date.

NOISE

Turning to noise, its types may be grouped as induced from (1) propeller, (2) machinery, (3) fluid flow, and (4) electrical components. Noise-level predictions in various ship locations appear more tractable if sound-power levels are known. Here again, noise associated with propeller-induced vibration presents a problem because of the difficulty of predicting the source sound-power level. Paper "N" demonstrated practical design methods for predicting noise intensity generated from sound-rated machinery in the surrounding space and at remote locations in the ship. With reference to main propulsion machinery and auxiliary units, a consensus developed that excessive noise levels in working spaces and accommodations originating from such machinery are often the result of neglect in the design stage, as remedial measures can be taken at minimum expense if dealt with during design development.

Although several noise criteria have appeared from various regulatory bodies including the Occupational Safety and Health Administration (OSHA), the lack of uniformity and a generally recognized standard of acceptable noise levels in the various inhabited spaces makes it difficult to decide on limits to include in shipbuilding specifications.

CONCLUSIONS

Papers presented in this Symposium dealt with all of the problem areas set forth above. Important contributions advancing the state of the art are discussed in the body of this report. Some of these are highlighted as follows.

Paper "H" demonstrated beneficial effects of partial stern tunnels and fins on wake homogenuity with attending reductions in propeller-induced vibrations. It also demonstrated a model technique for measuring effective wake by simulating flow sucking action of the propeller with diffusers.

Paper "I" reorganized and greatly simplified a previously developed formula for calculating propeller-induced vibratory hull surface forces for ships with sterns which are broad and flat aft.

Paper "J" clarified the interrelationship of hull and machinery and the problems of compatibility with respect to deflections and vibrations, taking into account correct and incorrect propeller shaft and diesel engine crankshaft alignment procedures.

Paper "K" identified computer programs useful in prediction of propeller-induced forces and moments acting on the hull surface and propeller together with programs available to calculate hull and shafting responses.

Paper "L" illustrated large-scale vibration analysis performed by the American Bureau of Shipping (ABS) with the aid of finite-element computer models. It also discussed proposed feasible solutions to the springing problem.

Paper "M" demonstrated how dangerous natural frequencies of hull structure can be recognized so as to permit a reliable choice of the number of blades.

Paper "N" developed a model for predicting noise levels in various inhabited spaces aboard ship.

Paper "O" recorded typical values of engine-room noise attenuation using existing techniques applied to main reduction gear casings.

Paper "P" presented a practical procedure for preventive maintenance of machinery by monitoring vibration signatures.

Discussers added more case histories to those of paper "R" of success with highly skewed propellers in suppressing objectionable propeller-induced vibration without any propulsive efficiency loss. This device is becoming widely recognized for its beneficial effects. There was even the inference that it could become standard for original propeller installations. The crash astern condition was identified as imposing the heaviest blade stress for fixed-pitch highly skewed propellers.

RECOMMENDATIONS

Out of this Symposium came clear recommendations for further research and development. The key areas generally agreed to included needs for:

1. Prediction procedure and verification of damping.

2. Refinement and verification of procedures for calculating added mass.

3. Refinement and verification of procedures for determining effective wake harmonic content in way of the working propeller rather than the nominal wake with propeller removed.

4. Development of a better method for predicting propeller pressure field acting on the hull in a cavitating regime.

5. Extensive full-scale testing on different ship types with measurements as necessary to develop data banks to verify calculation and model test procedures and provide empirical coefficients to bridge gaps until uncertainties in the prediction methods are dispelled.

6. Much more extensive feedback for both vibration and noise conditions aboard operating ships to better coordinate research programs.

7. Uniform rational standards of acceptable noise exposure in inhabited spaces.

8. Machinery-manufacturer measurements and dissemination to shipbuilders of sound-power levels and structure borne acceleration levels of their products.

9. General simplication of design procedures to be set forth in a handbook which would enable a typical shipyard staff to make reliable checks on vibration and noise prior to fabrication. It should also be useful to naval architects and marine engineers in preliminary and contract design phases.

PART II INTRODUCTION AND TABULATION OF COMPLAINTS

The interagency Ship Structure Committee and the Society of Naval Architects and Marine Engineers jointly sponsored this Symposium to bring together representatives of the maritime community from the United States and from abroad to discuss all aspects of ship vibration, noise, and machinery/hull incompatibility. This report covers the contents of 18 papers delivered at the Symposium together with discussions and authors' closures.

COMPLAINTS

A good starting point begins with the chorus of complaints raised during the entire Symposium relating to disagreeable and costly consequences of excessive vibration and noise encountered on many recently built ships. These complaints together with related costs serve to emphasize the seriousness of the problem which persists in spite of improved technology development from several decades of intensive research.

Several factors account for the apparent inability of designers and shipbuilders to positively guarantee quiet ships in the average situation. Today's ships are larger, more powerful, and more likely to have all accommodations and navigating station aft over the main machinery and closer to the propeller, the two principal sources of vibrations and noise. Moreover, with mechanized handling of unitized cargoes, containerships and roll on/roll off ships as well as bulk carriers spend most of their time at sea thereby exposing the ship's crew to the sea environment over a greater percentage of their working lives.

Apart from direct problems of subdueing responses incident to the growth in machinery power per se, larger ships confront stiffer propeller shafting interacting with machinery mounted on structure inherently more limber with longer athwartship spans of the supporting floors.

Evidence that these factors continue to produce serious consequences may be appreciated from the following list of complaints culled from papers "B" and "E" which represent shipowner and maritime labor viewpoints and as also appeared throughout other papers and the discussions. These are essentially excerpts with some paraphrasing and are presented in two tables as follows:

TABLE 1 - SHIP DAMAGE DUE TO VIBRATION

1. Owner required to shock mount or relocate radars, navigation gear, communication units and helm.

2. Cracked welds and fractured strength members in underdeck area in wheelhouse.

3. Some navigation equipment and aids rendered useless at various speeds.

4. Frequent repairs of wheelhouse equipment necessary.

5. Vessel personnel diverted to attend to accommodation noise maintenance and repair broken water pipes.

6. P.V.C. pipes failed throughout vessel due to vibration. Corrections were expensive.

7. Factory shock mounts of navigation and communication equipment failed after less than normal life expectancy. Replacements delayed-not shelf items.

8. Radar mast vibration sufficient to break loose and drop antenna.

9. Most of the aft peak framing of a relatively new 40,000 tons tanker found in a heap at the bottom of the tank.

10. Epidemic of economizer element fractures.

11. Lignum vitae stern bearings pounded out.

12. Spring-mounted condenser vibrated in blade frequency with about two inches amplitude.

13. Major failures have occurred in switchboards.

14. Alarm panels falsely activated.

15. Pump foundations loosened.

16. Cracked pump castings.

17. Steering gear

    a. Broken pipe connections.

    b. Broken hydraulic line hangars.

    c. Motor coupling locking device allowed to disengage with uncontrolled 90° course change.

    d. Broken or loosened electrical connections of power and control units.

18. Falling objects in engine room, i.e. from machine screws to handwheels.

19. Hammer and vibration in cargo system so severe that all flanges needed tightening after each discharge.

20. Steering-gear hydraulic piping required repair several times during one voyage.

21. Fractured hangars and brackets of steam and drain lines during course of one voyage.

22. Deck cargo handling gear vibration so severe as to compel ship speed reduction.

23. Blown gaskets.

24. Damaged gears.

25. Frequent calibration settings.

26. Steam gland leakage.

27. Diesel engine crankshaft bearings crazed.

28. Main reduction gear: cracked teeth, pitting, spalling and undercutting of teeth.

29. Engine-room fire due to excessive shaft vibration.

30. Cracked hull plating.

31. Cracked propeller ducts.

32. Fire damage insurance jeopardized based on alleged negligent failure to correct vibration in fuel lines (From Noonan's closure paper "A")?

TABLE 2 - PERSONNEL DISCOMFORT DUE TO VIBRATION AND NOISE

1. Loose panelling, cable racks, pipe supports created high noise levels while vibrating.

2. Mess room aft vibrated coffee out of cups.

3. Cantilevered bridge wings flopping, excited wheel house and prevented writing on chart table.

4. Excessive vibration in forward part of vessel.

5. Midship deckhouse vibration interfered with deck officers' sleep.

6. Deckhouse vibration caused engineers to resign.

7. Ship's lounge unusable.

8. Saloon uncomfortable.

9. Vibration concentrated in deck officers' quarters.

10. Dropping and hoisting anchor woke up off-duty personnel.

11. Forced draft fans caused house to vibrate.

12. Major engine-room noise complaints - main engine reduction gears, generator, air compressors and fuel oil transfer pumps. Verbal communication impossible in machinery space of motorships.

13. Hearing loss.

14. Noisy main feed pumps.

15. Steam reducing station created unbearable noise.

16. Cargo pump noise and vibration transmitted into quarters.

17. Dry-cargo winch noise and vibration.

18. Lack of sound insulation. Normal conversations heard through bulkhead.

19. Reduced personnel effectiveness.

20. Less than OSHA protection now available. The Coast Guard has not followed the lead of OSHA in promulgating regulations.

21. Effect of noise, vibration, and ship's motion are cumulative.

22. Vibration of a writing desk made handwriting difficult.

23. Dinner things jingle.

24. Cabin made a restless impression.

PART III VIBRATION

GENERAL

Paper "A" contained a comprehensive overview of the state of the art in principles of shipboard vibration and included comparative guidelines from several sources for habitability criterion. It then illustrated the successful application of current technology to recent designs of a destroyer and a LNG carrier. Volcy, in his discussion of paper "A", described procedures used successfully on LNG carrier EL PASO structural design of thrust block foundation, double bottoms and superstructure and also the propeller shaft vibration calculations which were equally successful in avoiding longitudinal and lateral vibration problems in service. Holden's valuable discussion of paper "K" presented a European contribution of Det norske Veritas research work carried out during the last decade featuring:

1. Preliminary stage design procedure based on data from 72 ships. Figures 6-8. Regression analysis showed good correlation between:

    a. Pressure impulses and

        1. clearances

        2. propeller

        3. wake field

        4. static pressure

    b. Superstructure vibration level and

        1. hull forces

        2. aft draft

2. Final design procedures, illustrated by Figure 10, covered theoretical methods and computer programs. As stated in paper "A", continuing research and development center on three basic avenues of attack; namely, (1) source strength prediction and reduction, (2) response prediction and detuning, and (3) improved alignment of structure and shafting.

Paper "C" provided succinct definitions of shipboard vibrations due to different sources as follows:

"Springing - Steady-state 2-noded vertical vibratory response of the main hull girder induced by short waves.

Slamming - Transient response of the main hull girder forced by impact with oncoming waves.

Propeller - Vibration of the main hull girder, local structure,
Induced   shafting and/or machinery due to alternating forces generated by the propeller.

Machinery - Vibratory response of a piece of machinery and/or the hull girder or local structure due to alternating forces produced in a machinery component."

Clearly, the predominant source relates to the propeller. Of necessity, it rotates in an uneven wake and, therefore, interacts with hull and machinery with impulses at blade frequency and to a lesser extent at higher harmonics of blade frequency. Because of its generally overriding importance, the Symposium devoted a large portion of its attention to this subject. However, in lesser proportion, the Symposium also dealt with the ship's interaction with waves, misalignment of shafting, and vibration monitoring techniques for preventive maintenance.

No one rebutted the observation that because of time pressure, complexity, and high cost of analysis during ship design stage, many shipowners and shipbuilders forgo analysis hoping that it will prove unnecessary. It was refreshing to hear Mr. Haskell say that his company fully appreciates the risks involved and was prepared to underwrite whatever costs were necessary to exploit all available technology in designs of new ships. Paper "F" underscored the wisdom of this approach by highlighting the serious economic consequences to ship operators of excessive vibration. Dashnaw identified the significant cost areas and presented an economic analysis of data provided in the report entitled "Cost of Ship Vibration Problems" by the Center for Maritime Studies of Webb Institute of Naval Architecture and from other sources. In his discussion of paper "B", Noonan noted that one must pay the price for necessary studies, design calculations, and shipbuilder responsibility. It is wishful thinking on an owner's part to expect a shipbuilder to undertake thorough vibration analysis in response to a loose specification such as requirement for "good vibration characteristics."

PROPELLER-INDUCED VIBRATION

Source Strength Reduction

The Symposium concerned itself mainly with three principal factors affecting forces and moments acting on the hull and propeller due to uneven wake; namely, (1) modifying or reducing nonuniformity in the wake, (2) the effect of propeller-blade cavitation on vibration forces, and (3) reducing vibratory forces by propeller-blade skew. Papers dealing with these factors contributed to procedures aimed at optimizing their influences.

Wake Optimization

Paper "A" illustrated stern profiles of three models tested for an LNG carrier with different stern configurations. Calculated results based on measured wakes demonstrated superiority of the open transom type over conventional and modified Hogner types as presented in Table I showing reduced unsteadiness of torque and thrust as well as bearing forces.

Paper "D" described, but did not illustrate, three different sterns for single-screw LNG ships which were model tested with varying amounts of deadwood cutaway. One was fitted with a bulbous stern. By installing fins over the propeller to the selected stern, tests showed that wake was homogenized with an added benefit of improved propulsive efficiency and ship speed.

Paper "H" illustrated half-body plans and stern profiles of seven designs for LNG ships together with model-scale measurements of wake and vertical pressure forces. These data showed that the application of partial stern tunnels leads to significant reductions in pressure fluctuations. This is due to the wake field becoming more homogeneous. The same result was obtained with a containership model illustrated in Figures 3 and 4 showing attainment of more homogeneous nominal wake pattern and reduction of dynamical cavitation behavior.

Paper "H" illustrated experiments performed on a Great Lakes carrier model showing the beneficial effects of partial stern tunnel on effective wake field, i.e. with propeller in place and running as simulated by a diffuser. Photographs of the model fitted with tufts demonstrated notable improvement in stream line flow into the propeller. Plots of wake measurements giving circumferential distribution of axial velocity components confirmed significant reduction of wake variation.

Propeller-Blade Cavitation Minimization and Blade Skew

Designers have recently learned that blade cavitation bubble build up and collapse concurrent with blade rotation through high-wake regions cause up to tenfold magnification of variable pressure field force acting on the stern in the vicinity and even some distance away from the propeller. Paper "K" states: "Since the growth and decay of a volume radiate pressure much more effectively than moving a volume from one place to another or introducing a flow from a source to a sink, the pressures from a small cavitation change can be large."

Paper "I" demonstrated by means of model tests the value of partial tunnel on a Great Lakes carrier. Before changing to partial tunnel type, severe pressure field force fluctuations augmented by propeller-blade cavitation contrasted sharply with the improvement obtained after installation. Further improvement was obtained by substituting a highly skewed propeller for one with zero skew.

Paper "Q" showed multiple frames of high-speed photography taking series pictures in a variable pressure water tunnel of four propellers having skews ranging from 0° to 72° and turning in a simulated wake. These pictures confirmed effects of skew on minimization of cavitation as predicted from model-scale unsteady force measurements made on a disc overhead of the propeller in the tunnel. In his view, the contributions from the non-cavitating propeller and the contribution from cavitation should be judged separately and then added together.

As to mechanism of the beneficial effects of skew on reduction of cavitation-induced forces, Bjorheden attributed it to more favorable tangential distribution of cavitation rather than smaller maximum extent of cavitation. This explanation is consistent with Hylarides' comment in

paper "H": "It is especially because of the attenuation in the explosive character of the cavities that a striking reduction in the vibration level is obtained."

Boswell's discussion of paper "R" depicted three actual propellers installed on different ships. He clarified options available to highly skewed propeller designers in the choice of degrees of skew and the radial concentration of skew. As against positive aspects of highly skewed propellers, paper "R" delved into higher first costs of designing, manufacturing and installing them. Hammer discussed economic trade-offs in four possible situation scenarios with generally favorable outcomes to highly skewed propellers. In these studies he took into account the higher costs together with probabilities that objectionable vibration would not necessarily be present with zero-skewed propellers. He also accounted for delay and cost of acquiring a highly skewed propeller if the need were indicated on trials. He further considered the costs of other remedial measures if taken instead.

Volcy's discussion of paper "R" provoked an apparent disagreement with the authors as to the efficacy of highly skewed propellers. Volcy contended that detuning was a preferable procedure to installation of highly skewed propellers for suppressing vibration in resonant structure because resonance may magnify response by a factor of 10. The 65 percent reduction attainable with highly skewed propellers would still leave an intolerable magnification factor of 3.5. However, the authors' closure cited effects of actual installations of highly skewed propellers which did in fact reduce responses by 65 percent and thereby produced acceptable vibration levels. Perhaps both parties could agree that magnification of responses of propeller shafting and gearing systems by a factor of 3.5 to torque and thrust variations may be intolerable, and hence selection of the number of blades should be made to detune away from shafting and gearing systems natural frequencies at the steady-state full-power normal operating condition, whereas detuning may not be necessary in most structural situations after installation of highly skewed propellers. In fact, in some situations involving major structure, detuning may not be practical or achieveable. Another factor relative to the two differing views may be in Hylarides' observation in paper "H" that: "In the range of blade frequency at service speed, the structure is not subject to strong magnification..." Whereas the opposite is true with regard to propeller shafting and gearing systems.

Predictions of Propeller-Induced Forces and Moments

Everyone agreed that the hydrodynamics of propeller-induced forces and moments acting on the hull and propeller remain in the domain of exceeding complex and still not fully understood phenomena. As Hylarides described it in paper "H": "It is a multi-component multi-related problem." The state of the art is such that important discrepancies exist between calculated, experimental, and full-scale values especially in propeller-blade cavitating regimes. As an example of such discrepancies, Noonan in his discussion of paper "D", indicated tests in the laboratory showed inception of cavitation at about 100 RPM, whereas aboard ship it took place at 60 RPM. Nevertheless, Bjorheden, in his discussion on this same paper endorsed model testing, imperfect though it may be, as it is always better to do some testing than doing nothing at all.

It is well known that propeller-induced forces are categorized in two groups; (1) the bearing forces - forces transmitted directly to the ship through the propeller and shaft, and (2) hull pressure forces - forces transmitted to the surface of the stern by the unsteady pressure field of the propeller. Their physical attributes and manner of propagation in the form of hull and machinery system vibration are described in paper "A" pages A-11 through A-13 and need not be repeated here. Suffice it to say that a major concern dealt with in this Symposium pertained to improving methods for predicting the magnitude and phase of these forces and moments during the ship design stage rather than suffering after the fact when the ship is already built.

Hull Pressure Forces

Paper "C" Table 2 referenced three computational methods for predicting propeller-induced surface forces.

Paper "K" recited historical development of computational procedures and then noted difficulties in their application. The authors assign 90 percent probability that the pressures can be predicted between 65 percent and 150 percent of the correct value for the non-cavitating condition. When cavitation is present but not excessive from the viewpoint of durability and efficiency, the accuracy of hull force predictions would probably be such that 80 percent would lie between 50 percent and 200 percent of the correct value. Currently available computer programs are described and in one case priced as to user fee at $5,000.

Paper "L" identified and described input to ABS/SURFORCE as a program which can calculate propeller-induced vibratory forces. It required knowledge of hull geometry, ship speed, hull wake, propeller geometry, RPM, and cavitation characteristics. Table I furnishes calculation results for a tanker from which a comparison of the magnitude and phase angle of the bearing forces with the surface forces can be made. Cavitation is accounted for by insertion of an additional theoretical thickness effect. Paper "K" characterizes this procedure as "unsound" because the pressures due to cavitation are generated by another mechanism other than that responsible for the pressure generated in the non-cavitating case.

According to paper "H", the lower harmonics of the wake field are mainly responsible for the hull pressure excitations, whereas for propeller-shaft excitations only some of the higher components are important. For this reason Hylarides concluded that both excitation systems are independent of each other. Prof. Vorus, in his discussion. took issue with this concept and indicated that except for the narrowest stern counters, the same harmonics of the wake field are predominant in the two-force systems. Although the zeroth harmonic dominates the induced pressure field, due to phasing, it integrates to near zero in all but cases of very narrow stern counter. The surface force is produced by the higher harmonic pressures of blade number which are directly associated with the corresponding wake harmonics. Therefore, he found it incorrect to state, in general, that propeller-induced hull surface forces depend most strongly on wake harmonics low relative to blade number. Hylarides' closure conceded Vorus' view to be correct for non-cavitating or moderately cavitating propellers. However, as soon as the dynamical behavior of the cavitation becomes important, the hull pressure fluctuation changes without any effect on the propeller-shaft excitations, proving for this and other reasons, that both excitation systems are independent of each other.

In his discussion of paper "I" Mr. Stiansen noted that Prof. Vorus' formula (2) for calculating the unsteady propeller-induced forces acting on the surface of a ship has been successfully applied to a Great Lakes ore carrier. It was found to be a reliable tool for calculating propeller-induced surface forces. However, the authors of paper "I" recognized that improvements could be made to make it more general and more adequately deal with the existence of propeller-blade cavitation. Accordingly, in an analysis and mathematical treatment they presented the reorganized formula as follows:

$$
\begin{aligned}
F_{in} = \frac{N}{2}\int_{r_1=r_h}^{r_t} \Bigg[& \sum_{\nu=0}^{nN} \left(\vec{K}_{nN-\nu} \cdot \vec{v}_{i\nu} - \rho i n N \omega Q_{nN-\nu} \phi_{i\nu} e^{i\nu\alpha t} \right) \\
& + \sum_{\nu=nN}^{\infty} \left(\vec{K}_{\nu-nN} \cdot \vec{v}_{i\nu} - \rho i n N \omega Q_{\nu-nN} \phi_{i\nu} e^{i\nu\alpha t} \right) \\
& + \sum_{\nu=0}^{\infty} \left(\vec{K}_{\nu+nN} \cdot \bar{\vec{v}}_{i\nu} - \rho i n N \omega Q_{\nu+nN} \bar{\phi}_{i\nu} e^{-i\nu\alpha t} \right) \Bigg] dr_1
\end{aligned}
\tag{2}
$$

Where:

$\vec{K}$ = vector representing blade loading and thickness at harmonic as denoted by subscript

F_{in} = amplitude of the n^{th} blade rate harmonic of hull surface force in direction i

N = number of propeller blades

r_1 = radius to propeller blade point in propeller coordinate system

r_h = propeller hub radius

r_t = propeller tip radius

n = blade rate harmonic order

ν = subscript denoting harmonic order

$\vec{v}_{i\nu}$ = ν^{th} harmonic of hull-induced velocity field in propeller disc propeller blade local cylindrical coordinate

ρ = water density

i = subscript defining direction of excitation force

ω = angular velocity of propeller

Q_m = m^{th} harmonic of the first time derivative of the blade cross-sectional area where m = subscript denoting harmonic order. The blade is considered a "pseudo-blade" composed of the material blade plus any attached cavitation

ϕi = hull-induced potential field in propeller disc

$\phi i\nu$ = ν^{th} harmonic of ϕ

d = local draft of hull in vertical plane of propeller

Inspired by Breslin's prior finding that the net vertical force (bearing force plus surface force) corresponding to a propeller operating beneath an infinite flat plate is zero, the authors simplified (2) for the case of ships with broad flat sterns aft. For an approximate evaluation of the cavitating condition, (2) becomes:

$$F_{3n}^{C} = \rho i n N^2 \omega \dot{V}_{nN} \left(\sqrt{z_o^2+\ell^2} - z_o\right) \qquad (18a)$$

Where:

F^{C}_{3n} = amplitudes of cavitating vertical hull surface forces

Z_o = vertical distance between water and propeller horizontal centerplane

1 = semi-beam of hull in propeller plane

$\dot{V}_{nN}$ = complex amplitude of the n^{th} blade rate harmonic of the cavity volume variation

For the non-cavitating condition (2) becomes:

$$F_{3n}^{NC} = -T_n \, v^{*}_{30x} - iF^{-}_{vbn} v^{*}_{31\theta} + iF^{+}_{vbn} \bar{v}^{*}_{31\theta} \qquad (27)$$

Where:

F^{NC}_{3n} = amplitude of non-cavitating vertical hull surface forces

T_n = n^{th} blade rate harmonic of alternating thrust

V^{*}_{30x}, V^{*}_{31} = hull-induced velocity harmonics at .7 propeller radius

The authors then illustrated comparative non-cavitating vertical surface-force predictions calculated by (2) and (27) and other approximations (flat plate formula) for four ship types. Three have relatively broad flat sterns and a containership has a counter relatively narrow to propeller diameter. Vertical surface forces are also illustrated in the cavitating condition as calculated by (2) and (18a) and other approximations for a Great Lakes ore carrier with and without a partial tunnel. The beneficial effects of the latter are evident. The authors concluded: "The approximate formulas developed

for calculating propeller-induced vertical hull surface forces are reasonably valid, at least in a relative sense, for sterns which are broad and flat aft, typical of open strut or transom stern ships. The formulas are not valid for ships whose counter is narrow relative to propeller diameter."

Vorus' and Breslin's discussions of paper "Q" expressed surprise jointly with the authors that the forces at second and third-order blade frequency are so small relative to those of the first order. As an attempted explanation, Breslin developed a theory and a suggested procedure for its evaluation. He would determine by integrating the theoretical pressures over Kerwin's disc if substantial selfannuling of the pressures due to harmonic variations with space angle γ occurs to produce reduced forces at second and third order of blade rate. The authors' closure described further experiments with a preliminary set of pressure measurements over the region of the force measurement disc. While the phase of the pressure varied over the disc, the changes were very similar for both the fundamental and twice-blade frequency components, thus confirming Vorus' and Breslin's theoretical conclusions. The results also showed that the peak pressure is to starboard of centerline similar to results obtained by Sasajima and Hoshino as reported in their discussion. In response to Rutgerson's discussion which suggested that signal averaging may have masked higher harmonic effects, the author's closure described their subsequent pressure measurements in which the signals were processed by a spectrum analyzer set at 1 H_Z band width. The relative amplitude of the harmonics continued similar to those obtained with the signal averager which implied a high degree of repeatability of each cavity collapse. Noonan's closure of paper "A" stated that measured second and third-order harmonic hull-pressure forces amounted to only 5% of first order for the LNG ship when fitted with fins. Without fins the second and third order were 34 percent and 6 percent respectively.

Brown's discussion of paper "Q" suggested a reverse technique for measurement of the transfer function of force to cavity volume velocity. The idea is to ocillate the disc and measure the resulting pressure in the propeller plane. Better yet would be employment of the water tunnel itself as part of the transducer measuring cavity volume velocity by exciting the tunnel with a shaker. The authors' closure promised further experimentation to validate the latter method.

Bearing Force and Moment Predictions

Paper "A" described and diagrammed the six components of bearing force and moment affecting the propulsion system directly due to non-uniformity of the wake in the propeller disc, i.e. longitudinal, torsional, lateral and vertical forces and moments: "The alternating blade frequency thrust of the bearing forces provides the principal excitation to the propulsion system in the longitudinal mode, while the blade frequency torque constitutes the principal excitation to the propulsion system in the torsional mode. The blade frequency vertical bearing force, when vectorially combined with the blade frequency vertical pressure force provides the total vertical force which excites the hull in the vertical direction. Similarly, the horizontal bearing forces, when combined with the blade frequency horizontal pressure forces, provide the major contributions for exciting the hull in the horizontal direction."

In his discussion, Hylarides objected from a mechanical viewpoint against vertorial combination of vertical propeller-shaft force and the vectorial hull pressure force: "Since these two excitations apply at different locations of the investigated sturcture and since the bearing force does not apply at the stern bearing but at the propeller, such a combination is inadmissible. With present knowledge and possibilities of detailed calculations I think we can better attempt to consider both excitations on their own merits:

- with respect to hull girder vibrations and so affecting the shaft vibrations

- with respect to the shaft vibrations so that via the bearings the hull is excited."

Noonan's closure paper "A" agreed with Hylarides that the number of blades should not be selected with respect to the strong wake components when evaluating a completed design or trying to salvage a new design. However, for an approach to new construction, the stern configuration would be optimized to minimize the hull pressure forces prior to development of design details.

Table 1 of paper "C" listed source references and computation methods for propeller-induced bearing forces. Paper "K" recited the historical development of these procedures. It also described computer programs and gave the user price of one ($6,000). In general, the methods all require as input, information on the wake, propeller geometry, and shaft speed. Paper "M" describing dynamic loading on the propeller stated: "...its calculation implies that of the pressure distribution on the propeller blade during one full revolution for a given propeller geometry and a given wake field.

The traditional approach using the so-called lifting line theory has been modified into so-called lifting surface methods featuring a sounder mathematical basis, but still making extensive use of correction formulae and tables based on systematic model tests.

Although some of the assumptions introduced are quite restrictive, such methods give reasonably accurate values of the dynamic loading on the propeller. Integration of those yields the resulting forces and moments acting on the shafting system, which are introduced into the model at the nodes coinciding with the bearings." Thus, theoretically determined propeller forces and moments may be used effectively in design. This is due in part to the finding that propeller-blade cavitation is less serious in its effect on bearing forces than on surface forces.

Paper "H" developed a variable pressure tunnel model technique for predicting effective wake by means of diffusers used to simulate flow sucking action of the propeller. In Figure 14, it compared static transverse shaft force and moment and the first harmonic amplitudes of the three components of the fluctuating shaft force and moment for a tanker and an LNG carrier from results of calculations based on nominal wake with results of calculations based on effective wake and results of measurements. Although some discrepancies still remain, the improvement in correlation of forces and moments from effective wake and as measured is evident.

In general there was no disputing that force and moment predictions from experimental model tests measurements are more reliable than calculations.

Occasionally, as with recent LNG carrier construction, time and money were available for extensive model experimentation during the design stage. Although opportunities for such testing arise infrequently, they are valuable for comparison with calculations and provide means for introduction of empirical factors to improve accuracy of calculated predictions.

Unsteady Forces Acting on the Rudder

In a discussion on paper "A", Mr. Hadler pointed out that there is a third mechanism exciting ship vibration which the Symposium did not cover. The hydrodynamic propeller-induced unsteady forces acting on the rudder can be serious depending to some extent on rudder location. Although this topic remained beyond the purview of this Symposium, it was noted as a subject deserving attention. Noonan's closure agreed and pointed to a number of other problems related to hydrodynamic excitation including rudder vibration related to "toe in" angle of twin rudders, strut vibration, increased propeller forces caused by bossings on twin screw vessels, propeller singing, etc.

Hull Structure Response

As paper "M" stated: "Calculations for predicting propeller-induced ship vibrations at the design stage involve the determination of four essential factors: rigidity, mass, damping, and excitation." Having estimated the propeller-induced unsteady forces and moments acting on the hull and propeller, the response of the hull girder can be estimated from mathematical models by the inclusion of estimates of added mass and damping. First, the resonant frequencies corresponding to vibration modes are calculated from the stiffness and mass characteristics of the system. The response is then computed taking into account added mass and damping. Paper "A" portrayed the state of the art with descriptions of the Timoshenko free-beam method which is useful particularly in preliminary design, but subject to limitations as Ward mentioned in his discussion of this paper. Paper "A" then described the finite-element method which can only be used in the more advanced stages of design after principal scantlings have been tentatively selected.

Paper "C" referenced and described computerized structural analysis programs currently usable. The ship-oriented programs model the ship as a beam and allow damping and bouyancy to be modeled. They also allow any number of subsystems to be attached to the hull and can analyze vertical and coupled horizontal-torsional vibration. The better known programs allow detailed simulation of the three-dimensional modeling of finite elements of the membrane, plate, and beam types.

However, these procedures, while dependable, are expensive and time consuming in developing input data. Paper "D" pointed out that "Modeling of the ship by a lumped mass-distributed stiffness type approach requires more experience derived skill but appears less time consuming." According to paper "D", "It would be appropriate to undertake a simple lumped mass-elastic axis analysis at an early stage, since hull inertia and shear area can easily be measured. By a simple, yet methodical, variation of parameters, the shipyard may then at least have some guidance on choices of structural continuity, house proportions, deck stiffness, and relative effect of vertical and horizontal exciting forces."

Paper "K" reviewed response calculation procedures together with related computer programs. It pointed out that the procedure recommended by the Ship Structure Committee project SR 240 is to assume that if the component substructures of the ship have suitable response characteristics, then the ship composed of these substructures will have suitable characteristics.

Following along this line, the forced response determination of the machinery space necessarily uses the finite-element method, preferably in a system compatible with that used for the complete ship so that the machinery space can be incorporated in the full F.E.M. model as a substructure. However, if the complete ship is modeled by a lumped mass-distributed stiffness approach, any convenient finite-element model can be used for the machinery space.

From paper "K": "It is desirable to make a study of the superstructure as a subsystem, since resonances in this region are a frequent cause of vibration troubles.... When the subsystems have been designed so that it is expected that they will be free of vibration resonances, it is time to make a vibration analysis of the complete ship. This analysis of the complete ship fulfills two important functions:

1. It checks and confirms the validity of the boundaries assumed for substructures.

2. By modeling the ship as a whole, it is possible, with the proper damping, to predict the vibration levels in all parts of the ship as a function of frequency.

Comparing these predictions with established acceptable levels allows an assessment of acceptability of the ship at a point in construction where corrections and changes to overcome serious difficulties can be determined and incorporated in the design."

Volcy's discussion of paper "K" furnished a description of a Bureau Veritas computer program available for hull steel work and machinery vibration analysis.

Paper "L" described the finite-element method application to the hull using fine-mesh grids for the afterbody and superstructure of an 1100 feet oil carrier, an ecological tanker, and a Great Lakes bulk carrier as performed by ABS. Calculated amplitudes at numerous locations throughout the ships are tabulated for the first ten modes in full load and ballast conditions. The beneficial effects of partial tunnels in reducing amplitudes are evident in Tables VII and VIII. On a matter of calculation technique, Stiansen disagreed with discussers Skaar and Smogeli who attached little interest to a single-point excitation force and would prefer presentation of the exciting force as force per unit length along the ship. As against this, Stiansen noted that if any area of the ship a short distance away from the hull bottom above the propeller, the response for a concentrated or distributed surface force is practically identical if proper modeling is used.

Stiansen dealt with the effect of bouyancy of the water on the ship in vibration prediction calculations by stimulating vertical springs where stiffness are equivalent to bouyancy effects at corresponding ship stations. Reed's and Burnside's discussion cited McGoldrick as authority

for neglecting bouyancy without influencing the results for vertical bending excessively. However, Stiansen's closure referred to McGoldrick's work as unaided by computer availability which today makes such calculations relatively easy and contributes significant accuracy to the final results.

Added Mass

Paper "L" described the ABS procedure for calculating added mass and its distribution based on linearized ideal fluid theory. Figure 5, showing a typical added mass distribution along the length of a medium size tanker, resembles the underbody sectional area curve to a remarkable degree. Tabulated value for added mass of an 1100 feet oil carrier is shown to be of the same order of magnitude as the mass of the ship.

Reed's discussion of paper "L" cited his attempt to use the authors' method for estimating added mass to be "very unhappy." As also mentioned in his comment on paper "M", he is looking into the treatment of water inertia in terms of the pressure field set up by a vibrating piston in a rigid wall. The authors of paper "M" noted that Prof. Webster at University of California at Berkeley developed a similar approach using a sink source technique which accounts for section deformation.

Paper "M" illustrated and described in detail the estimation of vibration frequency and amplitudes of the engine room and aft peak of a 122,000 cubic meter LNG carrier by finite-element modeling. The structural modeling is conventional. However, the authors introduced a novel procedure for calculating added mass for which the claimed superior accuracy, especially in the higher vibration modes corresponding to blade frequency and twice-blade frequency. In this method, they employed a three-dimensional finite-element discretization by using fluid elements with curved boundaries so as to allow a perfect fit with the immersed part of the hull. As set forth in the paper, "The complete model of the liquid domain includes six sub-domains and is entirely described using 372 twenty-node isoparametric fluid elements, 2229 nodes with one-degree-of-freedom per node, and 283 nodes in contact with the hull. Since only one-degree-of-freedom (the dynamic pressure) has to be considered, the computer time... is negligible...." In his discussion, Skaar suggested that added mass determined by this method may not be more correct than estimated by simple strip theory, since only the global deformations are of interest in the model.

Skaar further noted and the authors agreed that to obtain the best possible coupling between water and structure, all 283 nodes and the corresponding high number of degrees of freedom should be included in the eigenvalvue calculation. Reed's and Burnside's closure of paper "K" faults the use of finite-element analysis to represent added mass on the basis of improper assignment of water inertia on an arbitrary basis using Lewis' values (applicable to long cylinders) to the multinoded vibration of a ship bottom structure. However, in his discussion of paper "M", Reed found that the use of a finite-element grid in the water, accurately represented the water inertia in the authors' mathematical model of the ship. The authors' closure cited their comparisons of values of natural frequencies obtained using the conventional Lewis approach and the fluid finite-element approach which showed excellent agreement for the first few modes of the hull girder with descrepancies increasing with number of modes.

Damping

Damping presents much more of an enigma than addes mass. According to paper "M"... the present knowledge we have of damping phonomena arising in complex structures cannot be satisfactorily modeled, as there appears to be a nearly complete lack of knowledge concerning this topic." The Symposium generally concurred although Chang, in his discussion, forecast enlightenment from Ship Structure Committee planned experiments which will isolate and measure components of damping forces. The authors of paper "M" agreed with Chang that a series of well planned experiments can solve the problem of damping in ship vibration.

Paper "L" noted: "... it is generally assumed that energy is dissipated by the following processes:

1. Structural damping

2. Cargo damping

3. Water friction

4. Pressure-waves generation

5. Surface-waves generation

6. Ship forward speed."

At propeller-blade frequencies and higher, the effects of surface waves and ship forward speed are small and can be neglected. For practical purposes, the effect of the other processes can be lumped together under the name of "internal damping" which presently must be treated empirically. Empirical damping factors are given in papers "A", "L", and "M". Paper "M" described full-scale vibration excitation and measurements on two ship deckhouses using Bureau Veritas Model E2000 generator, a powerful exciter. Comparison of measured acceleration and calculated amplitude of response showed significant differences attributed to lack of knowledge due to damping. Obviously, computations of displacements and accelerations at specified points of a ship cannot be rigorous regardless of precision of the exciting forces. However, comparison of the results of calculations and measurements can lead to data banks of damping coefficients which in the long run should prove useful for improving accuracy of the empiricism.

Machinery System Longitudinal Vibration

Paper "A" noted that the resonant magnification factor for longitudinal vibratory response of the main propulsion system to unsteady propeller thrust can vary between 9 and 12. Thus, resonance with alternating thrust at blade frequency and multiples thereof need to be avoided in the thrust load-carrying structure. As noted in paper "K", this is partially accomplished by providing sufficient stiffness in the thrust bearing foundation to assure that a shaft longitudinal vibration does not fall in the operating range. Static stiffness calculations during preliminary design can be made by representing the foundation and bottom as a combination of frustrums of wedges and beams as described in Reference (24) paper "K". It also can be done by finite-element method or by representing the machinery-space double bottom as an anisotropic plate.

With respect to shaft longitudinal vibration analysis, the solution can be calculated on the basis of distributed shaft mass. However, Stiansen in his discussion referred to experience with a few nodes placed along the shaft representing the weight of the shaft as lumped masses. He indicated a similar degree of accuracy could be obtained with this method particularly if only the fundamental mode is of primary interest. Reed's and Burnside's closure to paper "K" agreed, but noted that with the computer program handling the mathematics, the saving in data input time and the reduction in probability of error gives an advantage to the distributed mass system.

Since the thrust bearing foundation is tied to bottom structure, longitudinal vibration of the shafting will excite engine-room vibration. Magnification occurs if, as it often happens, the natural frequency of the bottom structure is not far removed from propeller-blade frequency. Paper "K" described computer programs designed to analyze unsteady longitudinal shaft forces at blade frequency and response of the machinery space. Mr. Stiasen asserted that unsteady forces at twice-blade frequency and higher are significant and suggested that finite-element models can deal with these higher frequencies.

Machinery System Lateral Vibration

Paper "K" discussed procedures to avoid hull structure resonance with propeller-induced force and moment excitations through the shaft bearings about axes normal to the rotational axis. Initially, the shaft is assumed simply supported rigidly at the bearings. As the design is developed, the effects of hull flexibility are considered. The paper described computer programs available for this purpose using the finite-element method. It also listed design changes which can be made, if needed, to avoid resonance with the shaft natural frequencies and with the supporting structure natural frequency. Because of practical considerations, the analysis methods presented in paper "K" are feasible only for the range of the fundamental blade frequency. Stiansen and Lane both questioned this, but as explained in Reed's and Burnside's closure in considerable depth, the limitation is not inherent in the finite-element analysis but in the economics of the use of finite-element analysis.

According to the authors of paper "K", the symmetry or lack of it in the bearing foundations would affect the directionality of two fundamental modes of shaft vibration (i.e. vertical and horizontal for the symmetrical case). Stiansen, in his comments, questioned this, suggesting that symmetry or lack of it may be immaterial. As can be inferred from his comment, the basic ship structural support, i.e. transverses and double bottom are vertical and horizontal. Thus their stiffnesses would be represented in the analytical model by two orthagonally placed springs say one vertical and one horizontal. The authors' agreed to the extent that the shaft support is symmetrical about a vertical axis. Otherwise, it must be represented by a matrix with off diagonal terms in the vertical and horizontal directions. As will be discussed in a later section under hull-machinery incompatibility, the vibratory response of the shaft within the bearing can exceed the bearing clearances with adverse consequences if the alignment is not good.

Machinery System Torsional Vibration

Propeller-induced unsteady torques at blade frequency transmitted through the shaft must be investigated to avoid resonance with the propulsion system

natural frequency and also to quantify the peak torque for gear tooth design. While these aspects are of the utmost importance, current technology appears to cope adequately, such that papers given in this Symposium barely touched on the subject.

Hull - Machinery Compatibility

Incompatibility problems between hull and machinery involved interrelated factors of misalignment, deflections, thermal expansion, and vibration. Again, the propeller transmits blade frequency unsteady forces and moments via shaft and bearings. This is the principal source of vibration affecting the main propulsion system. It is unique to ships, whereas other sources, which at times can be more important, such as dynamic unbalance or bent shafting, commonly arise in land installations and have already been understood and treated exhaustively.

Paper "J" profusely illustrated two types of damages due to incompatibility; namely; damages to bearings and gear teeth. As mentioned earlier, under the topic of Machinery Systems Lateral Vibration, the vibratory response of shafting within bearings can exceed bearing clearances with adverse consequences, especially if misalignment exists to begin with.

Paper "J" discussed the influence of the hull girder deflection and local hull structure deformation in way of the engine room and illustrated finite-element meshes of engine-room double bottoms used in the analysis. The author also indicated the influence of hull structure deformation on distribution of shaft bearing reactions. He then demonstrated the effect of thrust bearing foundation stiffness on tilting and misalignment of main reduction bull gear. Also, the shaft stiffness and location of line shaft bearings are shown to influence bull gear alignment. Finally, he discussed procedures to achieve the all critical "rational" or correct shaft alignment. Thereafter, he devoted the latter part of his paper to similar discussions of the "rational" alignment of crankshafts in large diesel prime movers.

WAVE-INDUCED HULL VIBRATION - SPRINGING

As defined in paper "C", springing is steady-state 2-nodal vertical vibratory response of the main hull girder induced by short waves. The long narrow shallow depth Great Lakes ore carriers are particularly prone to springing because their low natural hull frequencies synchronize at times with steady-state periods of encounter with short waves. The responses in terms of bending moment and deflection are significant and, therefore, must be considered in design.

Paper "L" reported on current research involving both analytical and full-scale experimentation. It described the computer program SPRINGSEA, a special type of random analysis, used by ABS (Paper "L" Reference 14). It was developed based on the integrated analysis of the hull dynamic response to wave excitation utilizing an energy spectrum and RMS values. "The program calculates the low-frequency wave-induced and high-frequency springing responses, the band-width parameter which checks the applicability of the Rayleigh distribution (Paper "L" Reference 15) and the statistics of the combined bending moment and bending stresses."

With reference to an extensive test program carried out on a number of Great Lakes ore carriers, paper "L" went on to plot comparisons between results obtained analytically and those computed from recorded data using ships' short-term responses and then semi-long-term responses. For the short term, the comparison showed a trend of higher springing stress than low frequency wave-induced stress in the low and intermediate sea states. For the semi-long-term response, good agreement was achieved between extreme measured stress values with theoretical curves for two different ships. Some discrepancy was shown for a third ship (CORT) believed due to errors in observed wave heights. In general, the results of the comparisons were satisfactory. Stiansen agreed with Cojeen's discussion wherein Cojeen pointed out that the correlation presented in the paper represents only one phase of the correlation work covering maximum stress levels which are obviously biased towards high-stress levels and do not represent the actual probabilistic distribution. Other phases are currently being pursued.

The importance of springing can be appreciated from test results analysis which indicate that in the severe sea-state condition representing a probable design target (30 feet significant wave height) the increase in combined stress due to springing is about 6 percent for the CORT.

In his discussion, Cojeen provided additional information on the potential springing problem in vessels other than Great Lakes bulk carriers. His plots, showing natural hull frequencies of seven ocean going tankers vs. deadweight, illustrated how the increased size vessels with lower natural hull two-nodal vibration frequencies find themselves in resonance with wave-encounter frequency more readily than older smaller tankers. He also confirmed springing activity on high-speed commercial vessels having natural frequencies which facilitate their excitation by energy in the high-frequency range of the spectrum.

Cojeen suggested that it will take considerably more full-scale testing to validate the correlation of Figures 21 through 25 of paper "L" as between the analytical model predictions and full-scale measurements. He then provided a description of the instrumentation and the current full-scale research program being conducted on the Great Lakes bulk carrier STEWART J. CORT. The effort includes the simultaneous measurement of the encountered waves and midship stress, torsion and numerous responses. Since wave measurement has always been a source of error with prior similar endeavors, great interest attaches to performance of the two instruments aboard the CORT to measure waves which will be checked out against wave-rider bouys to be provided by NOAA.

Stavovy's discussion described the DWTNSRDC structural seaworthiness digital computer program ROSAS. The results of its use have correlated well with actual ship responses. The program simulates hull girder structural response including ship rigid and elastic body motion, bending moment and shear. Vibrating hull girder modes can also be determined.

VIBRATION SIGNATURE ANALYSIS AS A PREVENTIVE MAINTENANCE TOOL ABOARD SHIP

Under this topic,the Symposium dealt with an area somewhat unrelated to anything discussed up to this point. Paper "P" described the potential for the application of portable vibration amplitude meter and vibration analyzer for preventive maintenance of miscellaneous rotating machinery units aboard ship.

The first step provides base-line data against which future measurements can be compared. This is accomplished by taking a complete set of vibration meter readings and a set of vibration signatures with the analyzer. Thereafter, a schedule of periodic routine vibration meter checks, hopefully, will alert the engineer to any machinery mechanical problems before they become serious. In event the vibration meter reading indicates a mechanical problem in a machine, a new signature is obtained with the analyzer which will indicate the specific frequencies which have changed, thereby pinpointing the trouble source.

The author provided illustrations of comparative signatures taken with the ship at rest which indicated the need for preventive maintenance. However, when the ship is underway, propeller-induced hull vibration, and in some cases, adjacent machinery-induced vibration will complicate the signature. The author then illustrated procedures identifying the environmental vibration components of the signature which should be discarded in the analysis.

While not new, the author claimed that vibration detection is just now being recognized as a valuable and practical aid to decreased shipboard maintenance and improved ship availability. He also suggested that opening up machinery for inspection can be curtailed without loss of reliability of the strength of unchanged vibration readings.

Discussers were less enthusiastic. They pointed out that classification societies were slow to waive periodic openings for inspection based on good vibration measurements. Manufacturers and contractors appeared also to be reluctant to accept normal guide lines furnished by the supplier of the analyzer-equipment.

The need for a data bank correlating conditions found upon opening for inspection with prior vibration readings appeared necessary on an extensive scale to improve the confidence level of vibration signature analysis. Some of this is underway. Doubts were also expressed as to the capability of a hand-held meter to detect high-frequency vibrations associated with anti-friction bearings or to cope unaided by the analyzer with environmental vibratory effects.

Nevertheless, on balance, there was general agreement that the procedure has promising aspects. Riksheim, in his discussion, speaking for Det norske Veritas, indicated that special survey arrangements for particular machinery components are presently in force for 10 turbine powered ships. In its research program,Det norske Veritas is attempting to determine,by calculations, acceptable vibration levels for each component installed, taking into account fatigue, deflection (clearance between rotating and static parts) and bearing loads.

Det norske Veritas experience indicates the differences between vibration levels for identical components may be large without it being possible to attribute this to different characteristics of the components.

Tremayne's discussion of paper "P" described ongoing systematic data gathering in the form of vibration measurements for specific pieces of machinery. Hopefully, consistently close correlation will be obtained between conditions indicated by vibration monitoring and analysis and conditions actually found during open-up inspections. The ultimate goal is to furnish convincing evidence which will enable classification societies to accept vibration measurements in lieu of open-up inspections.

CONCLUSIONS

General	Source*
1. The design effort to reduce shipboard vibration has to be an evolutionary process that never stops at the contract design but carries into the detail design.	C Swensson CC
2. Vibration affects: fatigue in the structure, annoyance to crew, and working condition of equipment.	A Smogeli
3. The cost of damage from vibration is significantly under reported. Damage from vibration may have been a major contributor to some casualties.	F
4. The growth of ship size with high-powered propulsive plants have resulted in relatively increased flexibility of steel structure in combination with increased line shafting stiffness which lead to incompatibility between hull and machinery accompanied by vibration affecting machinery and hull.	J
5. In the past, not enough attention was paid to interaction between machinery and hull.	J
6. Of 41 ships measured by Det norske Veritas on sea trials, 24 had vibration troubles.	A Smogeli
7. Attaintment of quite high-powered ships with present technology requires extensive analysis, disciplined design and a measure of good luck.	D
8. Shipyards are not safe in adopting specifications with ship acceptance limits and guarantee penalties based on vibration performance. However, paper "A" illustrated application of current technology to a destroyer design and both papers "A" and "D" described similar vibration control designs for different LNG carrier types. All ships easily met stringent vibration criteria on trials. The authors conceded, nevertheless, that the good results may have been fortuitous to some degree.	D Noonan DC
9. Prediction of the actual vibration behavior of ship under service conditions is not possible at present.	M

*Note: Letter denotes paper from which conclusion was discussed. If a name appears, it is that of a discusser. Letter followed by C indicates author's closure.

		Source
10.	The correct evaluation of the stiffness of a ship is feasible. The causes of errors in vibration analysis are due to approximations involved in the calculation of the hydrodynamic forces, the excitation forces and the damping of the structure and cargo.	M Chang
11.	A complete survey of full-scale observations of propeller cavitation plus many comparisons of different ships full-scale with model results confirm the tendancy of model results to show a lesser extent of cavitation both with respect to RPM and angular positions than occurs full scale.	H Holden
12.	Model test measurements of propeller forces and moments are more reliable than calculated values. However, in preliminary design, theoretically determined propeller and hull forces and moments are useful and usually better than empirical data.	A D Bjorheden DC
13.	Cavitation effects are best assessed in a vacuum tank.	A
14.	Hull pressure forces can be measured in a cavitation tunnel as well as in a vacuum tank.	A Ward AC
15.	To determine whether or not the obtained excitation levels will lead to an acceptable vibration level, a comparison with similar existing ships has to be carried out or a response calculation has to be performed.	H
16.	Mathematical modeling can serve to estimate hull girder and main machinery responses by the application of propeller forces and moments and the inclusion of added mass and damping estimates.	A
17.	One of the most productive courses one can take in vessel design is one in which close attention is paid to structural continuity. See also Wu's discussion of paper "C" in which he graphically illustrated two similar sterns. One good and one bad from a vibration standpoint. Longitudinal side girders provided necessary stiffness for the good performer, whereas the bad performer did not have this longitudinal stiffness and continuity.	C Kline Wu
18.	There seems to be some reluctance on the part of shipbuilders and shipowners to fully employ the analytical techniques available. The objection of high cost should be insignificant if employment of complete analysis were limited to large high-powered ships, since provision of adequate propeller clearances and good flow into the propeller disc should suffice to insure freedom from objectionable vibration for low-powered ships.	K KC

	Source
19. Uncertainty of estimating procedures and lack of verification are the two main deterrents to the application of hull vibration analysis procedures.	C
20. There are many computer programs available for predicting the loading and response of a ship due to propeller-induced excitations.	K
21. Only a few specialists can claim sufficient familiarity with the assorted computer programs available so as to be qualified to direct the most cost-effective vibration analysis for a particular design.	K Dillon

Excitation - Propeller Forces - General

22. The primary effort towards controlling shipboard vibration should be directed at controlling exciting forces.	A
23. The principal forces relate to propeller-blade frequency or harmonics of propeller-blade frequency.	A
24. In preliminary design, shaft forces can be ignored and instead one can estimate the total force entering the hull based on inputs from model studies.	AC
25. In general, to keep propeller and hull excitations low, it is desirable to use many blades on the propeller. Holden's discussion disagreed because propeller cavitation normally gives the main contribution to surface forces. Reed's and Burnside's closure first rebutted this comment as inapplicable outside the cavitating range. However, they conceded that Mr. Holden's experience in predicting hull pressures from a cavitating propeller showed better accuracy than their own.	K Holden C
26. The beneficial influence of high blade skew on the propeller-induced vibratory forces was demonstrated for both noncavitating conditions.	Q Bjorheden
27. As to skewed propeller lower excitation level, a more favorable tangential distribution of cavitation resulting in lower time dependent fluctuations of the cavity volume contained within the critical region of the propeller disc plays an important role.	Q Bjorheden
28. Operating and model test results indicated that there is no noticeable speed penalty on ships fitted with highly skewed propellers. Model tests may miss target propeller RPM up to 5 percent.	R
29. Highly skewed propellers reduce overall ship vibration levels approximately 50 percent. Greatest improvement appears to coincide with resonant conditions where highly skewed propellers may reduce vibration levels 65 percent or more.	R

		Source
30.	A highly skewed propeller for a high-powered single screw cargo ship will cost up to $50,000 more than a conventional propeller.	R
31.	In case of resonance, the predominant factor being the dynamic magnification of the response, an important decrease of vibratory level cannot be expected solely due to adoption of a highly skewed propeller. Detuning is necessary either by changing number of blades or natural frequency of the vibrating structure. Hammer and McGinn disagree. See last paragraph under topic "Propeller Blade Cavitation Minimization and Blade Skew."	R Volcy RC
32.	Propeller design technology has not yet developed analytical or model techniques for determining optimum skew angles and/or distributions.	R
33.	Tests with three blades with 120° warp giving results of pressure fluctuation measurements compared with non skewed propeller at a shaft inclination of 12° showed no reduction of amplitudes.	A Hylarides
34.	Overall, the successful development of highly skewed propellers for merchant ships represents the single most important advance in propeller-induced ship vibration reduction technology within the past decade or perhaps within the last century.	R
35.	The highly skewed propeller is not a cure for all conditions. Therefore, continued vibration control research is warranted.	F
36.	Stern configurations should be optimized by testing several for a given design. This can be achieved by flow-visualization and self-propulsion tests.	A & L
37.	Design details of a given stern configuration, such as fin or partial tunnels, propeller clearances and rudder locations, can significantly influence the forces generated.	A
38.	To reduce the hull pressure fluctuations, good results can be obtained by replacing the propeller if it lacks certain refinements including: pitch reduction at the tip, greater blade area, increased skew.	HC
39.	Due to dominant role of the effective wake field, which still cannot be measured accurately, theoretical investigations, using nominal wake-field input are, therefore, susceptible to inaccuracies.	H
40.	Analytical procedures for the calculation of cavitation on propellers require the effective wake field as input. Theoretical studies at best serve as qualitative approach to this problem.	H

		Source
41.	It is easier to change propeller blade number than it is to change the dynamic characteristics of propeller shaft system or main substructures such as deckhouse. Therefore, the number of blades should not be selected with respect to strong wake-field components.	A Hylarides
42.	Vertical hull pressure force and propeller bearing force should not be combined vectorially since these two excitations apply at different locations of the investigated structure.	A Hylarides
43.	Forces and moments referred to axes normal to the rotational axis of the propeller shaft can probably be estimated to a 90 percent probability that the predicted value will lie between 75 percent and 140 percent of the correct value.	K

Propeller Surface Force Excitation

44.	In recent years, nearly all excessive vibration problems were solved by reducing the hull pressure forces and moments.	H
45.	According to Holden, European technology is ahead of U.S. technology in developing theoretical methods for calculating hull surface forces including effects of cavity volume fluctuations.	K Holden
46.	The present knowledge of propeller-induced hull pressures is far from satisfactory.	M
47.	For a given model wake field, the noncavitating propeller-generated hull-pressure amplitudes can be estimated for various propeller designs and clearances.	K
48.	Model pressure measurements require less expensive testing time than null-force methods.	KC
49.	The present programs can predict the excitation and response except in the analytical prediction of cavitation pressures on the hull surface which can be significant.	K
50.	Propeller-blade cavitation can magnify hull pressure forces by factors greater than ten to one.	A
51.	The presence of cavitation on the blades is not the dominant role in generation of pressure forces, rather it is the growth and collapse of the cavity that have the tremendous effect.	A
52.	The influence of cavitation on the propeller-induced pressure fluctuations should be judged separately from pressure pulses obtained in the non cavitating condition, and then the two should be added to each other.	Q Bjorheden QC

		Source
53.	Since induced pressures are proportional to the rate of change of cavity volume, both maximum extent and circumferential distribution must be considered.	QC
54.	In the non cavitating condition, hull surface pressures adjacent to the propeller tip can probably be estimated to a 90 percent probability that the predicted value will lie between 65 percent and 150 percent of the correct value.	K
55.	In the cavitating condition, the accuracy of hull force predictions would probably be such that 80 percent would lie between 50 percent and 200 percent of the correct value.	K
56.	Due to the simplicity of the Kerwin experiment,as further analyzed mathematically by Vorus in his discussion of paper "Q", this would certainly be a cheaper and more expedient method (and quite possibly more accurate one) than experiments on ship models for evaluating cavitation-induced hull surface force.	Q Vorus QC
57.	An important contribution of the calculation methods developed in paper "I" is the insight obtained with regard to the influence of propeller location relative to the stern and the shape of the hull over the propeller.	I Gray
58.	Experiments and theoretical investigations have both shown that hull surface forces are independent of the number of blades because propeller cavitation normally gives the main contribution to surface forces.	A Vorus
59.	For ducted propellers at blade frequency, the non cavitating part of the pressure is usually dominating due to small clearance, while at higher harmonics the contribution from the growth and decay of cavity bubbles will always predominate.	K Carlsen
60.	Constant RPM operation over a large speed range with controllable pitch propellers requires a propeller-blade design with large reserve against face cavitation to avoid blade erosion. This in turn inherently makes it difficult to avoid sheet cavitation on the suction side at full power and therefore aggravates surface forces causing vibration.	H Hagemar HC

Propeller Bearing Force Excitation

61.	Amplitude and phase of the propeller harmonic longitudinal forces and harmonic torque about the rotational axis can probably be estimated to a 90 percent probability that the predicted values will lie between 85 percent and 120 percent of the correct values.	K

		Source

Wave Excitation

		Source
62.	Significant hull girder bending may occur due to the excitation of the beam like low mode vibrations of the ship by the energy present in the corresponding frequency range of the sea spectrum.	L
63.	Large vessels operating in the Great Lakes represent a class of ships in which wave-induced vibrations are more important than in ocean going ships.	L
64.	The effect of springing on the combined stress depends on significant wave height and more importantly on the stage of sea development as characterized by the peak frequency of the wave spectrum. The increase on the CORT in deck stress ranged from about 10 percent in a fully developed sea with a significant wave height of 15 feet to 60 percent in a developing sea characterized by the same wave height.	L
65.	In a severe sea condition with a significant wave height of 30 feet which more closely approximates the design condition, the increase in combined stresses due to springing is about 6 percent. This is enough to affect the design section modulus.	L

Response

66.	Accuracy of response predictions can only be determined by comparing calculations with measurements. The comparison is best made on actual ships excited by a shaker. The best comparisons have been made on space vehicles (see reference 5 of Reed's and Burnsides closure).	KC
67.	As to response calculations, if the component substructures of the ship have suitable response characteristics, then the ship composed of the substructures will have suitable characteristics.	K
68.	Structural and/or mechanical resonances should be avoided in the operating speed range. Hull criticals and shaft RPM, propulsion system resonances and number of propeller blades must be considered.	A
69.	Detuning is an effective treatment of resonant vibration problems.	J
70.	Difficulties inherent to any full-scale measurement serve to partly explain poor correlations between calculations and measurements.	M
71.	Crew comfort has been improved on all ships fitted with highly skewed propellers to date.	R

Source

Hull Response

72. Existing empirical formulas for the estimation of natural hull frequencies are outdated for many of the new ship types and sizes. C

73. Although complex structural models yield a good approximation of the rigidity and the structural mass of the ship, there is, on the other hand, a lack of fundamental knowledge of added mass, damping, and excitation forces. This deficiency is being filled by simplified methods which are often questionable. M

74. The 20-station beam model is useful in preliminary design, whereas the finite-element method is best used for design evaluation of major substructures and propulsion systems. Also a finite-element model of the stern area in combination with a beam model of the forward portion of the ship is useful for detailed studies. A

75. Regarding added mass, fluid finite-element discretization makes possible a more accurate approach to the dynamic fluid-structure interaction. It is now possible to determine with suitable accuracy the set of natural modes of a ship and particularly of the afterbody in a frequency range (5-15 Hz) which was not accessible by calculation using conventional added mass. M

76. For local structure, damping is small and resonance must be avoided. H

77. At propeller blade frequency and higher, the hydrodynamic damping is very small compared to internal damping and can be neglected. L

78. In the range of blade frequency at service speed, the structure is not subject to strong magnification due to resonance; therefore, the hull girder response is, to a high degree, independent of frequency. H

79. Bouyancy springing can be safely neglected at propeller-blade frequencies. KC

80. Around frequency of the hull girder lower modes, the hydrodynamic damping is usually predominant. L

81. Comparison of the results of calculations with measurements of exciter-generated vibration yields, for a given excitation, useful value of damping coefficient.

82. The present knowledge of damping arising in complex structures cannot be satisfactorily modeled, as there appears to be complete lack of knowledge concerning this topic. M KC

		Source
83.	Prediction of potentially dangerous afterbody vibration modes can be accomplished by comparing the relative influence of the natural modes on the frequency response after introducing arbitrary but realistic damping and unitary excitation in the dynamic equation of motion (Paper "M" (1)).	M
84.	The fundamental mode of the superstructure may be calculated with sufficient accuracy with a reasonably sized F. E. M. model. However, to obtain the hull-induced resonant peaks and to calculate the vibration level, a complete afterbody model is necessary.	L Skaar Smogeli LC
85.	Athwartship and torsional vibrations can be severe and should be accounted for in any vibration analysis.	D Acker DC
86.	Although ships may have similar characteristics of hull girder lower modes and maximum response of afterbody structure at propeller-blade rate, many of the results, however, cannot be predicted without a realistic finite-element representation of the ship's afterbody. Special areas under investigation must be represented by a relatively fine-mesh model.	L
87.	The magnitude and phase angle of propeller bearing forces have an important effect on the response of the vessel's afterbody when combined with the surface forces.	L
88.	The combined calculation of responses of two sterns with equal bearing forces can be three times smaller for the stern without tunnel even though the calculated force is about eight times larger due to the effect of force phase angle.	L
89.	In order to correctly assess the values of all deformations of steel work needed for rational shaft alignment, it is necessary to build up a correct finite-element-method model of the aft part of the engine room.	J
90.	Generally it will be found that the natural frequency of the bottom structure in the machinery space will not be far removed from the propeller-blade frequency.	K
91.	Rigid vinyl modeling can be used to determine the best structural arrangement of machinery space to minimize large deformations.	J Stavovy JC
92.	Response of the structure to a single vertical force may allow potentially dangerous frequencies to be recognized so as to permit a reliable choice of the number of propeller blades.	M

		Source
93.	Cracks in structure are found close to the main excitation sources, e.g. aft peak over propeller.	A Smogeli
94.	Vibration level causing crew annoyance is approximately 1/10th of the level causing cracks in the structure.	A Smogeli
95.	When measuring vibration aboard ship, a deck on its own is not a sufficiently typical vibrating object. Measurements should be made in addition of bulkheads, table tops, and equivalent important objects within a living space.	A Janssen

Machinery Response

96.	The number of propeller blades is set primarily by the frequency of the shafting and propeller in longitudinal vibration.	K
97.	The simplest but not very good procedure for predicting the natural frequency of the shafting and propeller is to predict on the basis of a one-degree-of-freedom system consisting of the propeller and water inertia plus a portion of the shaft weight vibrating against the stiffness of the thrust bearing and foundation.	K
98.	An improved procedure is to model the propeller and shaft as a series of concentrated masses and elastic elements and use a Holzer process for frequency distribution.	K
99.	If a computer is used, a more accurate result can be obtained if the shaft is represented by a continuous mass and elasticity distribution. ABS experience indicates that by placing only a few nodes along the shaft representing the weight of the shaft as lumped masses it would suffice to lead to a similar degree of accuracy.	K Stiansen
100.	The problem of hull/machinery compatibility requires cooperation between naval architects and marine engineers. Trouble-free operation can be obtained in a propulsion system if professionals from both disciplines agree on and work effectively toward maximum deflections within allowable at each point of support in the overall machinery system including turbines, gears, and shaft bearings.	J Budd
101.	Good structural continuity, strength and stiffness, particularly with regard to foundations of major machinery, is critical in order to avoid vibration problems.	C Kling CC

		Source
102.	The resonant response of the tailshaft, its hammering against the forward stern tube bearing, can be mitigated by correct alignment of line shafting which will detune the resonator and cancel the dynamic amplification of the response of the shafting.	J
103.	Highly skewed fixed-blade propeller-blade strength design criteria are governed by astern rotation when absorbing about 50 percent of engine power.	R Bjorheden

Vibration Signature Analysis for Preventative Maintenance

104.	Preventative maintenance based on analysis of machinery unit vibration signature will promote improved ship reliability leading to cost reductions.	P
105.	The owner who specifies maximum vibration tolerances of his machinery at purchase and who follows the final installation with additional requirements for minimum vibration readings as a criterion for acceptability has given himself an edge in assuring machinery reliability.	P
106.	Vibration signature analysis provides positive indication of specific faults such as unbalance, misalignment, or defective bearings. It also gives a quantitative number which can be used to evaluate the severity of the defect.	P
107.	By using vibration signature analysis along with traditional inputs of temperature, pressure, sound and touch, the engineer is in a much better position to minimize operational problems.	P
108.	A shipboard program can be implemented with two portable instruments: a hand-held vibration meter and a vibration analyzer/XY recorder which provides graphic signatures for each machine to pinpoint defects detected by the vibration meter.	P
109.	When environmental vibration occurs aboard ship, it tends to mask a machine's self-generated vibrations and thus obscure the true mechanical condition of the machine.	P
110.	Techniques described in paper "P" are available to separate environmental vibration signatures from machine signature.	P
111.	ABS is not yet in a position to accept vibration signature analysis as a means of survey in lieu of opening up and examining due to uncertainties as to consistent reproducibility of the technique and lack of guidance criteria.	P Blanding

		Source
112.	Det norske Veritas is experimenting with 10 turbine-powered ships. They conclude that it is necessary for the marine engineers aboard ship to become more involved.	P Ricksheim Mogen
113.	Vibration signatures are repeatable within about 10 percent, if measured under the same conditions each time.	PC
114.	Rolling up to 10° has little effect on the vibration signature.	PC
115.	First and second harmonics of blade passage are below the frequency region of interest and, therefore, cause little problem with the hand-held meter.	PC
116.	The technique is applicable to reciprocating machines but requires more analysis and care in setting conditions.	PC

RECOMMENDATIONS

1.	Prepare a loose-leaf text book covering criteria, analysis procedures and experimental methods which are now spread over many sources. New information should be added from time-to-time and outdated information should be removed.	C
2.	Survey the behavior of ships and their propulsive plants with sufficient measurements to check correlation with prior analysis and provide data bank and empirical coefficients to bridge gaps until uncertainties in the prediction methods are dispelled. This is particularly relevant to damping and pressures on the hull surface.	J
3.	Undertake thorough ship-instrumentation programs involving measurements of wake, hull pressures, shafting bearing forces and moments, structural response and observation of cavitation, all as aids to removing uncertainties in analysis methods.	D
4.	Shipyards themselves should do much more in the way of vibration measurements because experience is the one thing which will prevent them from being too wrong. These should be coordinated industry-wide efforts as is done in Europe.	D Ward DC
5.	Because of the important successes achieved to date with highly skewed propellers, further development appears of great urgency to (1) enable the average designer to develop good designs and (2) to avoid compromising the effective work done to date as the result of future failures due to poor design. Systematic series testing is needed to develop design curves covering ranges of advance coefficient, amount of skew, number of blades	R Dillon

		Source
	and power levels. The same holds for propeller-blade strength with special attention to reverse rotation when backing.	
6.	Encourage shipowners to install at least one highly skewed propeller on a ship in each fleet.	B Hammer
7.	For more unconventional propeller systems, for instance, highly skewed propeller, wide-bladed propeller, and propeller/duct systems and operation at low advance rates ($J \rightarrow 0$), there is need for complete unsteady lifting surface methods, including tip vortex separation and cavity flow.	K Holden
8.	It is necessary to have the detailed force and phase angle distribution along the hull as well as the hydrodynamic shaft forces as input to forced-vibration analysis.	K Holden
9.	At a later stage of design procedure, there is a need for more advanced method to take into account details about the propeller design e.g. blade profiles, shape of leading edge, skew, rake, the detailed local wake-field distribution, and to be able to analyze more unconventional hull/propeller configurations.	K Holden
10.	Extend paper "Q" experiments to a less idealized wake having peak value of 0.6 to 0.7 near the twelve o'clock position.	Q Sasajima & Hoshino & Hadler
11.	Conduct further measurements of the effective wake for several kinds of ship forms by using diffusers to validate the method.	H Sasajima/ Hoshino
12.	Analytical prediction of cavitation requires continued research.	K
13.	Discrepancies between RPMs at cavitation inception as predicted in the laboratory and as occurs aboard ship needs investigation.	D Noonan
14.	There is need to investigate propeller-blade pressure side cavitation and propeller/hull efficiency.	K Holden
15.	More extensive data are needed which the designer may use to decide whether or not cavitation-induced vibration is likely to be a major problem.	Q
16.	The authors of paper "I" should include a definition of the function of equation 18a for a flat plate of finite width and infinite length to make it applicable for a systematic variation of hull form as corrections to this flat plate assumption.	I Gray

		Source
17.	A larger effort than Hylarides' presentation on effectiveness of wake-adapting stern tunnels is needed. Such data might be provided by parametric studies where several parent hull forms are tested with a series of appendages (tunnels, vortex generators, steps, fins, etc.) A parameter characterizing the degree of wake nonuniformity along with power increments would be evaluated for each variation.	H Vorus
18.	Further investigations are necessary to make clear the mechanism of the effect of the partial tunnel fins.	H Takehuma Sasajima Noonan HC
19.	A very important improvement in correlation between model and full-scale statistical data will be obtained if the full-scale data would be derived via normalized measuring procedures as proposed by ISO.	HC
20.	Although paper "I" developed a simplication of a general formulation for calculation of propeller-induced hull surface forces, the practicing engineer would probably find it difficult to use. Further simplication is needed. Vorus agreed but also pointed out that full-flow data presented in the paper for four ships is recommended for first estimates of vertical hull surface forces for similar ships. Parametric presentation useful to designers will be possible after more computations using the formula are conducted. See also conclusion 16.	I Stiansen
21.	The formulas developed in paper "I" are useful for designing a propeller for minimum vibration for a given set of hull lines, since the hull nominal wake is invariant, so that the need for new model tests with each variation can be avoided provided propeller cavitation is not an important factor.	IC
22.	Measurements of the pressure distribution of the second blade frequency component on a flat plate at each phase of propeller rotation produced small values. It is doubtful, therefore, that tip vortex cavitation is always responsible for the second force. Since it depends on cavitation patterns, further study is necessary.	
23.	The contradiction between higher harmonic blade cavitation surface forces as predicted by paper "Q" experiments and as obtained from pressure measurements over model sterns is confusing and may have led to improper propeller design with harmful vibration as a consequence in an actual instance. It is necessary to undertake research into the importance of higher blade frequency harmonics upon ship vibration problems.	Q Hadler

		Source
24.	Evaluate the importance of the effect of forward speed on the mass matrix, damping matrix, and stiffness matrix in the equation of motion (1) of paper "M". Since they are important in seakeeping theory, they must also be important in vibration analysis.	M Chang
25.	Vibration analysts should look to models such as NAVSEC's CASDAC concomitant with development of structural design for the volume of structural configuration data required.	C Billingsly
26.	Further research is needed in damping and its distribution over the various nodal modes of the ship structure. According to Reed's and Burnside's discussion, generally damping in the form of a percentage elastic energy loss per vibration as a complex modulus of elasticity matches the vibration of damping with frequency. More good predictions are needed for comparison with good detailed measurements. Stiansen's closure was skeptical of Reed's and Burnside's suggestion and faults it for going well beyond present capabilities of the largest computer programs without adding to the correct physical modeling of the phenomenon.	L KC Reed Burnside
27.	Further research is needed for a more accurate derivation of the added mass as a function of frequency. Because water inertias are such a large part of the vibrating masses, it is important that accurate values for water inertia be used.	L LC Reed Burnside
28.	More research is needed of both fundamental and applied nature so as to confirm the validity of the linear viscous model, i.e. F.E.M. discretization of water surrounding the hull. In case it is proven to be a good assumption, it will be necessary to characterize values of damping coefficient in various modes for ships of a comparable nature.	M
29.	A set of standards should be agreed to among machinery manufacturers defining the maximum allowable relative deflection at each point of support in the overall machinery system including turbines, gears, and shaft bearings.	J
30.	Needed research now underway in springing: Fatigue studies Basic research in springing excitation mechanism Theoretical and experimental research on method of combination Development of stress monitoring instrumentation Investigation of the rationale for hull flexibility Plan for measurement of hull damping	L Cojeen
31.	The future development for shipboard use of vibration as a preventative maintenance tool must be concentrated in two areas:	P Harrison

		Source
	a. Development of guideline for shipboard use realizing the outside influences unique to a vessel which will be accepted throughout the industry. See also Maxham's and Hardaway's discussion of paper "P" in which they recommended and gave samples of a fault identification table and a severity classification chart.	
	b. Formal pilot programs presented to and carried out for classification societies which will allow their rules to be altered utilizing vibratory records and current machinery signatures to approve the class of this machinery without opening for inspection.	
32.	Deterministically based acceptable levels of vibration should be obtained for each component installed.	P Riksheim
33.	The effect of propeller-induced unsteady pressure forces on the rudder needs further study.	A Hadler
34.	Custodians of hull-response programs should conduct and publish a "sensitivity analysis" for a typical ship(s). The mean response amplitude and phase vs. frequency per unit of force applied for each of the sundry propeller forces are needed. Such an analysis will allow an understanding of the relationship between the force vector (gravity, surface, thickness and cavitation) and response, which is not evident from published analysis.	K Gray
35.	Because of the reluctance of shipowners and ship operators to discuss vibration-related costs in an open forum, an informal means of accumulating vibration-related costs should be undertaken as part of the process of identifying worthwhile research programs.	FC

PART IV NOISE

DISCUSSION

Paper "A" described the state of the art in noise control aboard ships. The author reviewed several sources of noise criteria including those promulgated by OSHA. He noted the lack of uniformity and consistency between various criteria, and he then developed recommendations for various shipboard spaces (Figure 10 Proposed Airborne Noise Limits). Figure 10 plotted curves for each type of space on coordinates of octave band center frequency Hz vs. octave band level - decibles.

In his discussion of the paper, Buiten supported the use of such noise rating curves which conform to practices in Holland although with some variation in limits as he pointed out.

Buiten and Janssen, in their discussion of paper "C", presented the state of the art of noise level predictions in accommodation spaces in motor ships as developed in Holland. According to the discussers, relatively simple and reliable procedures have been used successfully during the design stage for more than 100 seagoing motor ships since 1967. The discussers listed valuable reference materials.

Paper "Q" cited the escalating financial consequences of noise to the U.S. Navy as the result of OSHA involvement. The author noted the rate of payment to shipyard workers for loss of hearing increased from 10 million to 40 million dollars per year within the three year period 1973-76. He then developed a readily understandable explanation of the OSHA noise criteria for permissible noise exposure.

The Symposium developed a consensus which recognized that within a practical range, compliance with various noise exposure criteria, including OSHA's, will not prevent all hearing loss claims. The problem centers on acceptance of noise level limits and exposure periods which will reduce probability of hearing loss to a tolerable extent for the general population.

As to analysis during design, paper "A" described the source-path-receiver approach to noise level prediction in three steps. These are: (1) determination of the sound power level, (2) estimation of the amount of attenuation in the paths between source and receiver, and (3) estimation of the sound power level in the receiving space. The author then described current technology applicable to these three steps. Paper "N" delved into this subject in greater detail as will be discussed below.

Propeller-induced vibration and noise go hand in hand. Therefore, what is good for minimizing vibration is also beneficial for noise abatement. According to paper "C", "Procedures for predicting noise levels in general do exist but more development and data are required before they become analysis tools. Consequently, the 'analysis' of noise is embodied in good design practice." The authors then summarized good design practice and guide lines which may include the following:

- Stern lines which allow good flow to the propeller and propeller clearance.

- Arrangement - segregation of machinery spaces from living spaces.

- Structural system that insures adequate primary and local stiffness and good continuity.

- Allowance of conservative weight margin for the foundations, particularly those for the main engine, gears and thrust bearing.

- Ventilation system with safe register velocity, baffler and large plenum chambers.

- Machinery resilient mounts.

- Shielding and insulation.

- Piping systems with safe flow velocities.

In paper "N", the author took a constructive approach to noise prediction and attenuation in ships. Noise control requires identification and knowledge of all significant noise sources. The principal ones are: "main and auxiliary engines, propeller, gear, casing and exhaust systems including funnel, various pumps, compressors, hydraulic systems and fan equipment including intakes and outlets." Nilsson's closure stated that the propeller, as an acoustical source, is being investigated in a joint effort between the Scandinavian countries. A first report concerning the response of the hull is to be published at the beginning of 1979.

Noise is either airborne, fluid borne, or structure borne. Methods for treating airborne sound are well known and can be reviewed in references (1) through (4) of paper "N". In rooms containing noise sources, the sound pressure level is almost entirely determined by airborne sound.

Papers on the subject of fluid borne noise are available from the National Fluid Power Association (NFPA) and the Society of Automotive Engineers (SAE).

Any mechanical force will induce structure borne sound the power of which is transmitted from a source through its connection to the foundation and is propagated in waves.

In Nilsson's words: "To make a prediction of resulting noise level in an accommodation space, the following quantities must be known:

- Source strength.

- Transmission properties of steel structures.

- Radiation properties of structures at the receiving end.

In order to define the strength of a source, it is, in general, sufficient to determine the velocity level perpendicular to the plating at the foundations of main and auxiliary engines, gear, pumps, etc. and in the hull plating above the propeller." As to machinery, he referred to semi-empirical formulae but indicated the reliability of the final results will be much improved if the input data are based on access to a data bank of actual measurements. He then

presented a short discussion on measures to attenuate source strength i.e. resiliency mountings, etc.

The propagation of structure-borne sound is a complex subject. Sound is transmitted as longitudinal, torsional, transverse, and flexural waves with coupling effects. Due to this complexity, propagation models must be simplified by assuming that one wave motion dominates and determines the power flow in the structure. After mentioning characteristics of the Statistical Energy Analysis method for propagation determination, and referencing reports on its application, the author turned to the Heckl analytical method for determination of vibration in grillages. He adapted it to analysis of superstructure noise propagation which considers the equilibrium of the bending moments of all joints in the structure from which the angular displacements can be solved as a function of the sound power input into the structure. The merit of this method was tested by comparing calculated and measured sound velocity levels for various decks on two tanker superstructures which showed reasonable agreement. As a by product, the author noted that although the exterior of the two superstructures were quite similar, the velocity levels differed appreciably. This reflected differences in plate dimensions and deck construction. It established the point that the dimensions of substructure must always be considered.

The author presented experimental results of model tests designed to appraise sound level attenuation by elastic superstructure mounting to the main deck and installation of damping layers successively on all deck levels of a model superstructure. The measured values were plotted to illustrate relative effectiveness. In one case where sound levels were calculated they agreed well with measured values.

The author also presented experimental results for Det norske Veritas conducted tests on noise radiation from bulkheads and decks on a full-scale mock up. It consisted of a section of a ship structure extending from the outer bulkhead to the casing which was excited by a vibrator. The resulting noise levels were measured and determined as functions of the velocity level of the steel deck.

Measurements were made on four floating floor constructions as described in the paper and plotted to indicate relative attenuation effectiveness. When floating floors were combined with elastic hangers for the ceiling the total noise level was decreased 20 db(A) without portholes or windows. Openings for windows can reduce the effectiveness of a floating accommodation system considerably.

Buiten, in his discussion, cited successful installations of resiliently mounted large diesel prime movers with decreased sound pressure levels in cabins of 10-15 dB. He suggested that opposition to such a solution is untenable.

With reference to floating floors described in paper "N" Buiten deflated their value for attenuating noise in the low-frequency range and concluded that they are relatively ineffectual for cabins situated above propellers of large sea going vessels, "This is due to limited additional mass and height of such floor construction which prohibit effective decoupling at lower frequencies."

Nilsson's closure stated that for a floating floor to be efficient in the low frequency range it is important that there be an acoustical mismatch between top and bottom plates. This is achieved if the top plate is thin. His closure also indicated that randomly chosen cabins on three different ships were measured with results as tabulated which showed that the total db-A level is determined by the noise level in the mid frequency region. However, particular attention should be paid to low-frequency noise.

Likewise, Buiten had reservations concerning the mathematical model of paper "N" for predicting the transfer of structure-borne sound when more extended sources than auxiliary diesel engines are considered e.g. propulsion diesel engines. As explained in detail in his discussion, Buiten found that: "Due to the strong coupled parallel systems, more modes may be expected at low frequencies than occur in the single-wave guide model. This would result in lower attenuation per deck for large sources than for small ones which would agree with our experience." Nevertheless, Nilsson's closure indicated that the model described in paper "N" has been used to calculate the propogation of structure-borne sound induced by main engines. The results have been satisfactory.

According to paper "O", acoustic complexities include:

° Environmental influences which can change measured sound levels many orders of magnitude.

° Non directional uniformity of sound radiation.

° Poor accuracy of available instrumentation.

° Effect of standing waves.

The author then indicated that noise source reduction of main propulsion gears, although optimized from a design standpoint, will not meet OSHA criteria. He, therefore, concluded that barriers and/or acoustic treatment are essential. Results of full-scale experiments with acoustical lagging of turbo generators and main reduction gear casings were presented together with a discussion of the proper location of microphones to measure the change in noise level due to acoustic lagging without interference from standing waves. Kugler, in his comments, confirmed the author's disparate results with lagging of turbo-generators and main reduction gears and pointed to the need for further investigation.

Paper "O" also noted that MarAd standard specifications do not recognize OSHA criteria. Wehr, in his discussion, considered MarAd's specifications, which seek to reduce noise levels of work and living spaces, to be more amenable to the development of equipment design standards. He noted that OSHA standards could be met merely by limiting the exposure time or by wearing hearing protection and thus could amount to no standard at all. Richardson commented somewhat differently indicating that MarAd specifications impose noise limits at the engine room console approximately equivalent to OSHA for an eight hour per day exposure. Speicher indicated in his discussion of paper "O" that revised MarAd specifications compatible with OSHA requirements should go to the printer in the near future.

Richardson also described a successful design procedure which assesses at an early stage all significant potential noise sources within a compartment.

Attainable noise limits are imposed on machinery equipment manufacturers and verified by testing at the plant, and acoustical treatment is applied as necessary.

CONCLUSIONS

Source

1. Noise control should begin in the ship design process. Post construction corrections are expensive. — A

2. The intensity of noise in the workplace affects health and safety of the seafarer. Thirty-three percent of Marine Engineers' Beneficial Association (MEBA) seafarers have sustained moderate hearing loss. — E

3. Noise exposure criteria show a large range of variability. Therefore, it is premature to establish a single number allowable equivalent noise level. Wehr disagreed, but admitted that the Lea criterion is difficult to measure and determine compliance. In any event, some exposure limit should be set. — G, Feldman, GC

4. Hearing loss above 2000 H_z is a significant impairment. — GC

5. The ISO 1996 noise rating (NR) curves provide a very reliable and well defined system for rating noise annoyance which provides a good amplitude frequency descriptor. When such a noise-amplitude frequency descriptor is combined with a vibration-amplitude frequency descriptor (given in Janssen's discussion of paper "A") a rating system is developed which shows good correlation between dose and subjective effect. — A, Janssen

6. The noise situation on board a ship is determined by the sum of noise contributed by the many and varied forms of noise sources. Effective control requires identification and knowledge of all significant noise sources. — N

7. There are only three ways to reduce noise: — G

   - Modify the noise output at the source.
   - Intercept the noise along its path from the source to the receiver.
   - Change the receiver's sensitivity e.g. hearing protection.

8. Total compliance with OSHA's current requirements will reduce hearing losses but will not prevent all hearing loss claims. — O, Wehr

9. There is little risk in 85-100 db-A exposure for short periods. It is the long-term cumulative effect which poses potential risk. — G, Feldman

10. The existing noise prediction programs should be considered as design guides rather than methods to calculate the actual noise levels in an accommodation space. — N

11. Despite certain shortcomings, the noise prediction methods existing today must be considered as of indespensable design assistance. — N

		Source
12.	There are two reasons to apply current technology during design: ° Design criteria may possibly be achieved. ° Arguments over responsibility may be avoided.	O
13.	Higher harmonics of propeller-induced surface forces are very often responsible for noise problems.	A Hylarides
14.	Structure-borne sound is directly induced by any mechanical force. The mechanical power transmitted from a source throughout connections to the foundation propagates into the structure.	N
15.	The type of mounting of each surface and its connection to the steel construction determine the velocity level between the deck and the radiating surface.	N
16.	Full-scale and model-scale experiments have indicated that the main noise power flow in the vertical direction in a ship structure is determined by the propagation of flexural waves in the plate elements.	N
17.	The sound velocity level of a deck is a function of the input power at the source, wave numbers, masses, losses, and dimensions of the plate elements of the structure.	N
18.	The sound power level in a room is a function of the acoustical power radiated into the room and also of the total absorption in the room.	N
19.	The power radiated by a structure excited by structure-borne sound is a function of the dimensions of the structure, radiation ratio, the coupling between the steel deck and the structure and the velocity level of the deck.	N
20.	Manufacturers of accommodation systems cannot supply radiation ratio nor coupling factor. All acoustical properties of accommodation systems should preferably be measured in situ aboard ship or in a laboratory in a special mock up.	N
21.	The concept of operating with some acceptable risk is fallacious, since inevitably there will be severe liabilities incurred by taking these risks.	
22.	The primary effort to control noise should be directed at the control of the propeller-induced hydrodynamic excitations.	N
23.	Noise limits should be imposed on vendor supplied machinery.	A

		Source
24.	Noise reduction treatments include: ° Ventilation silencers or duct treatment. ° Bulkhead and decking acoustic treatment. ° Machinery vibration isolation. ° Floating deck structures	A
25.	The gain obtained by reducing excitation, generally, is considerably larger than the gain that can be obtained by using acoustic material.	
26.	Poor quality control in the construction phase can nullify designed noise control. Airborne leakage paths and "shorting" of vibration isolators are common fabrication faults.	A
27.	The noise in a shipboard space may be predicted within about 5 to 10dbA for airborne noise when the source sound power levels are known. Otherwise, the degree of prediction is probably greater than 10dbA.	A
28.	When deciding which of the noise treatments to apply, the noise source levels must be known and whether they are airborne or structure borne.	A
29.	Buffer zones comprised of passageways and infrequently manner spaces provide considerable acoustic attenuation.	A
30.	The attenuation of structure borne sound is a function of frequency.	N
31.	No simple and general rule can be formulated concerning the attenuation of structure borne sound in a steel structure. The dimensions of the substructure must always be considered.	N
32.	The further away the measurement point is from the damped area the smaller is the effect of the damping layer.	N
33.	Damping layers on the vertical plate sections increase the attenuation of flexural waves more than longitudinal waves.	N
34.	The effect of damping layers on full-scale structures is considerably less than in the scale model.	N
35.	Openings for windows can reduce the effectiveness of a floating accommodation system considerably.	N

SOURCE

RECOMMENDATIONS

1. Give priority to machinery spaces for prevention of hearing damage and to ships' bridges for good communication. — G

2. Better planning is the most important thing needed. This must begin with the owner and the design agent and include: — O

   ° A better understanding of the many complexities of acoustics, including measurement of suppression.

   ° Acknowledgement of practical minimums in terms of practical limits at the source.

   ° Recognition of the probable need for barriers and/ or other noise-reduction techniques.

3. Regarding instrumentation, specifications should address themselves to the type, the calibration of, and the techniques in using sound-measuring equipment. — O

4. Naval architects should pay close attention to the developing field of research in infrasound (low frequency, sub-audible, 1-20 Hz vibrations) which has potential for degradation of operator efficiency. However, the authors have no knowledge of a procedure to treat the problem directly in the early stages of design. — C Billingsly CC

5. Decisive action is needed by the U.S.C.G. to meet the governmental mandate and establish the seafarer's safety and health standards. — E EC

6. Make available to the Coast Guard for use in rule making action twelve years of clinical data gathered by MEBA. — E Brown

7. Perform audiometric testing of exposed personnel at least annually. — G

8. Develop a comprehensive and accepted standard for noise criteria pertaining to the marine industry. Use a weighted equivalent noise criterion. Give consideration to noise levels recommended in paper "A" Figure 10. — O Wehr

9. Designate all areas which exceed 90dbA "CAUTION-HEARING damage area" including normally unmanned spaces which must be entered from time to time for inspection and maintenance purposes. — A AC

10. Include paper "A" Figure 10 noise level limits in shipbuilding specifications. — A

11. The concept of a 24 hour or one week noise level equivalent criterion appears to warrant study by the marine industry. — G McGinn

		Source
12.	Improve noise-prediction technology particularly with regard to structure-borne noise.	A
13.	Conduct additional full-scale verification studies aimed at improving the confidence level in the prediction process.	
14.	With respect to structure-borne noise some of the most important topics to be investigated are: ° Coupling between the main sources and the steel structure. ° Prediction models for the description of the power induced by engines and propeller. ° Description of the propagation of structure-borne sound in the double bottom. ° The total attenuation in a ship structure as function of the losses in the vertical and horizontal directions.	N
15.	The magnitude and direction of the energy flux of the various in plane waves remain to be investigated in structure borne noise.	N
16.	Encourage machinery vendors to measure airborne sound power levels and structure borne acceleration levels of their products.	A
17.	More acoustical data on typical sources should be collected and stored in data banks.	N

☆U.S. GOVERNMENT PRINTING OFFICE: 1979-311-586/250